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USE OF AN AMSLER WEAR TESTING MACHINE TO INVESTIGATE THE WEAR OF STEEL

Robert Clifford Doxey

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USE OF AN AMSLER WEAR TESTING MACHINE

TO INVESTIGATE THE WEAR OF STEEL

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Robert C. Doxey

USE OF AN AMSLER WEAR TESTING MACHINE
TO INVESTIGATE THE WEAR OF STEEL

by

Robert Clifford Doxey

Lieutenant, Unites States Navy

Submitted in partial fulfillment
of the requirements
for the degree of
MASTER OF SCIENCE
IN
MECHANICAL ENGINEERING

United States Naval Postgraduate School
Monterey, California

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This work is accepted as fulfilling
the thesis requirements for the degree of

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IN

MECHANICAL ENGINEERING

from the

United States Naval Postgraduate School

PREFACE

Wear occurs when metal to metal contact takes place between the mating surfaces of two machine parts in relative motion. In many service applications the useful lives of the parts are determined by the amount and type of wear occurring. The ability of a material to withstand the wearing process is termed the wear resistance and is greatly influenced by the conditions of service. The relative wear resistance of materials can be evaluated only in service or by the use of testing apparatus which duplicate the service conditions.

The objective of this thesis was to investigate the resistance to wear of a plain carbon steel using an Amsler Wear Testing Machine. The machine, one of several types specifically designed for the wear testing of metals, was recently acquired by the Postgraduate School, and the tests were the first to be conducted on the machine. Therefore, the project involved the installation of the machine as well as the actual testing. It was desired to learn if the machine was functioning properly. For that reason the tests were patterned after tests performed on an Amsler machine at the National Bureau of Standards by Mr. Samuel J. Rosenberg, Metallurgist.

The writer wishes to thank his advisor, Professor Roy W. Prowell of the United States Naval Postgraduate School, for encouragement and valuable assistance. Appreciation is extended to Mr. Rosenberg of the National Bureau of Standards and to Mr. Allen K. Schleicher for their interest and advice.

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CHAPTER I

INTRODUCTION

1. The need for wear resistance

That materials wear in use comes as a surprise to no one, for the phenomenon seems as natural as the rising of the sun or the passing of the seasons. We accept the fact that the children's new shoes, the new suit of clothes, and the shiny new automobile will be gradually destroyed by the destructive processes of wear. It has been said that

> nearly everything that moves causes and is exposed to wear, simultaneously bringing about and undergoing progressive destruction [1].*

Wear ranks with corrosion as a great destructive force to which metals are subjected. Gillett [2] has defined wear as

> the undesired change of dimensions in service resulting from pressure and sliding exerted by some other body.

When this dimensional change causes a machine part to fail to perform its designed function, the useful life of the part has ended. Obviously, any factor which can lessen the wearing rate and increase the service life of a part is a desirable factor indeed.

The ability of a metal to withstand the destructive forces of the wear process is termed its wear resistance. All other things being equal, any increase of wear resistance will reduce wear and increase the service life. Unfortunately, the property of wear resistance is not intrinsic. It can not be considered by itself and can not be measured and assigned a value as can the density. Wear resistance must be evaluated always in terms of the specific conditions of service. The relative wear resistance

* Numbers in brackets refer to the Bibliography on page 45.

of materials is best found by actual service use. This is usually impractical in a testing program, and a number of machines have been designed to test materials under various simulated service conditions.

2. Types of wear

With regard to metals a number of types of wear are found to exist, and the differences among the several classes of wear are not always sharp. Another confusing factor is the lack of a standard nomenclature. However, the tiny fraction of material lost in the wear process seems to be one factor that the various types have in common. Shidle [3] has estimated that a five ton truck, completely worn out, weighs only five pounds less than when new. Another writer [4] has called wear "the tremendous trifle".

The surfaces of mating parts are not smooth but consist of peaks and valleys in an irregular fashion. The various types of wear are explained in terms of the behavior of these asperities. Cutting wear occurs when a projection of one surface cuts off or breaks off an asperity of the other. The process might be said to be analogous to machining. Abrasion wear is a form of cutting wear. In this case loose particles between the surfaces perform the cutting. It is evident that the tips of surface asperities removed during cutting wear remain in the interface for a short time at least. They in turn cause abrasion wear. Therefore, we may consider cutting wear and abrasion wear to be in the same category.

As a second category, consider the effect of surfaces mating under a normal load. Opposing surface projections meet in some instances, and due to the very small areas involved the stress at a point of contact may be very large. Plastic flow occurs and very high instantaneous temperatures

are reached. A welding action occurs, and the terms galling, seizing, and scuffing are descriptive of the result. Pitting often occurs as a result of normal load and appears to be related to fatigue caused by repeated stressing. As the name implies, pitting results in surface pits. Way [5] conducted tests under rolling friction with heavy normal loads and reported that pitting occurred only when the rolls were lubricated. Spalling results in the separation of small flakes from the surface and is sometimes similar to pitting in appearance. However, fatigue is not involved.

3. The relationship of lubrication

Metal to metal contact of the mating surfaces of machine parts in relative motion can be prevented by hydrodynamic lubrication. The load is balanced by pressure developed within the lubricant which completely separates the surfaces. Friction is due to the viscosity of the lubricant alone. Although this is sometimes called thick-film lubrication, Kingsbury [6] obtained hydrodynamic lubrication with a film thickness of only 25 microinches. Mechanical wear does not occur as long as hydrodynamic lubrication is maintained.

When sliding speed decreases or the load increases, the film of lubricant becomes thinner. If hydrodynamic pressure fails to support the load, opposing high spots of the mating surfaces come together and deform plastically until the area is sufficient to support the load. If part of the load-supporting area is prevented from metallic contact by an adsorbed film, boundary lubrication exists. Bowden and Tabor [7] envisage complete support of the load by opposing surface asperities. A small part of the load-supporting area is in metal to metal contact.

The remainder of this load-supporting area is separated by adsorbed molecules of lubricant. Some wear occurs with boundary lubrication, but it is small. The type of wear involved is the galling or welding type, and the frictional force is the sum of the forces required to shear the junctions formed and the lubricant film.

As long as the shearing of the junctions is accomplished at the interface, the wear is gradual and not catastrophic. However, when the junction proves stronger than the base of the asperity, shear occurs at the base. Relatively large amounts of metal are plucked from the surface, and the wear is highly damaging. It is a function of the so-called extreme pressure lubricant or E. P. lubricant to provide a low-shear-strength material between the rubbing surfaces [6]. A chemical is added to the lubricant. This reacts at the surface of the metal to provide the low-shear-strength material. Surface metal is consumed in the process, but the wear is gradual rather than catastrophic. As before, the frictional force is the sum of the forces required to shear both the junctions formed and the film of lubricant.

Shaw and Macks [6] and Bowden and Tabor [7] are not in complete agreement insofar as boundary and E. P. lubrication are concerned. However, it is believed that the brief discussion above is a plausible one.

A fourth classification is dry or unlubricated friction. Bowden and Leben [8] observed unlubricated sliding to be a stick-slip process involving the welding type of wear. The forming and breaking of the junctions caused a violently fluctuating friction force. Bowden and Tabor [7] concluded that dry friction was the sum of the force required to shear the junctions and a plowing force required to displace metal from in front of

their slider. They further concluded that the frictional force remained proportional to the applied load only so long as the actual contact area increased in proportion to the load. Films of metals and films of oxides were found to eliminate this proportionality at low loads.

Extremely high coefficients of friction and excessive wear were observed for perfectly clean metals by these investigators. However, metals in service are never clean in this sense of the word. Adsorbed oxides and other impurities provide protection analagous to boundary lubrication, and the resulting frictional force and wear rate are much lower than would be expected for a clean metal.

4. Other factors involved in the wear process

The effect of hardness on wear was considered by Holm [9], who concluded that where the true area of contact was proportional to the applied load, the weight of the material worn away should be directly proportional to the applied load and to the distance traveled and inversely proportional to the hardness.

Surface finish can be very important in some instances. Lane [4] reported that extreme smoothness of cylinder bores resulted in increased galling of piston rings due to wear-in difficulties of the piston rings. Some degree of surface irregularities seems to aid in retaining lubricant. A process called superfinishing [10] was reported to result in greatly increased service lives for automotive parts. In this process a very thin layer of material is removed by a lapping operation performed at extremely light pressure. The operation supposedly removes material left in a highly stressed condition by machining and grinding operations.

An important factor in the wear resistance of mating surfaces is the

degree of affinity of the two metals for one another. Like metals tend to form the welds at the interface readily, and certain pairs of dissimilar metals have high score-resistance [6]. The welding action can be explained in terms of the high stresses and flashes of temperature during plastic flow, but evidently an adhesion phenomenon is involved also.

The oxidation of wearing surfaces influences the degree of wear considerably. One effect is preventing the actual contact area from increasing proportionally to load [7]. Loose oxide particles cause abrasive wear. In some cases oxides exert a protective influence upon a surface [11]. Fink [12] concluded that wear oxidation was a separate type of wear to be ranked with cutting and welding.

5. Wear testing

Inasmuch as many variables are involved in the wear process, no single testing machine can be used to evaluate the wear resistance of materials for all conditions of service. The conditions of service must be simulated for a test to have any meaning, and usually any attempt to exaggerate service conditions to shorten the testing cycle will introduce new variables [1].

Many different machines are on the market for testing the wear of metal against metal. Others simulate service applications of metal against a nonmetal or abrasive [1]. One of the oldest and most widely used testing machines is the Amsler Wear Testing Machine, which is described in detail in Appendix I.

Many investigators have reported the results of tests on the Amsler machine. Rosenberg [11] conducted an extensive investigation of the

variations in the wear resistance of plain carbon steels with increasing carbon contents, various heat treatments, and different conditions of loading. Fink [12] reported some rather astonishing results on his investigation of the effect of oxidation. Running steel against steel in air with one per cent slip and 50 kg. load, he obtained 0.1802 grams weight lost. He then conducted the same test in an atmosphere of nitrogen from which oxygen had been carefully removed. This time he measured 0.0000 grams of weight lost. Rosenberg [13] attempted duplication of Fink's work and obtained appreciable rates of wear on tests conducted in air, nitrogen, and hydrogen.

CHAPTER II

APPARATUS

1. The Amsler Wear Testing Machine

All tests were performed on an Amsler Wear Testing Machine [14,15]. This machine, manufactured by Alfred J. Amsler and Company, Schaffhouse, Switzerland, is designed for the wear testing of metals under a wide variety of test conditions. Fig. 1 is a photograph of the machine, and a detailed description of the machine, its accessories, and various configurations is offered in Appendix I.

In the author's tests cylindrical test pieces were mounted on the parallel shafts of the machine. These shafts were driven by a single electric motor through a system of gearing and rotated in opposite directions. The speeds were about 200 r.p.m. and 180 r.p.m. for the lower and upper shafts respectively. A known normal force was introduced along the line of contact of the specimens. Thus, in addition to rolling friction a tangential frictional force acted at the line of contact because of the speed difference. The upper specimen was caused to reciprocate axially, and therefore, a constantly varying, longitudinal frictional force was also introduced at the line of contact.

A Veeder counter was geared to the lower shaft and indicated revolutions of the lower specimen. A dynamometer, located between the lower shaft and the electric motor, indicated the torsional moment required to turn the lower specimen. This moment and an angular velocity proportional to that of the lower shaft were inputs to an integrating mechanism. Another counter on the integrator indicated the total frictional work

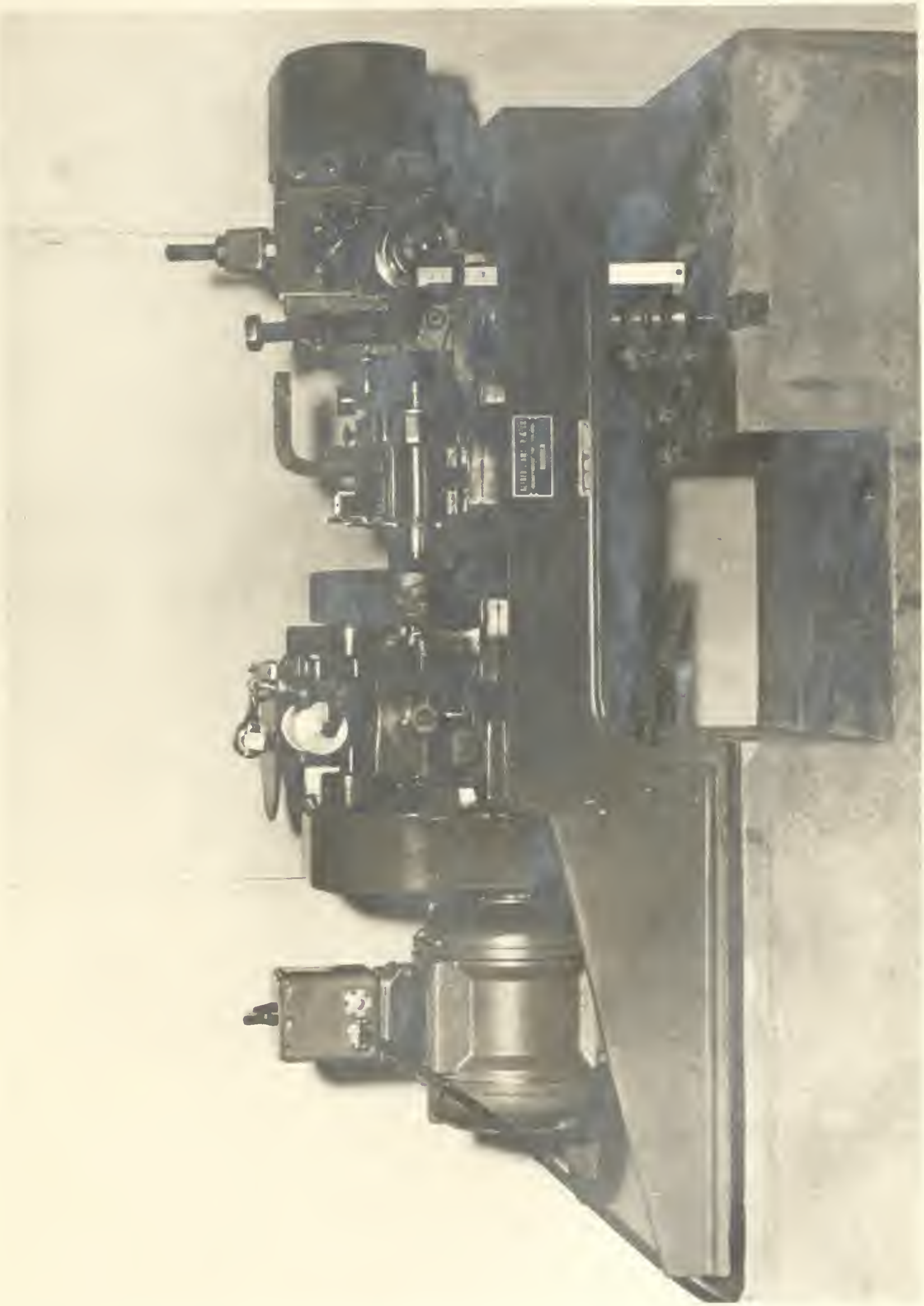


Fig. 1. The Amsler Wear Testing Machine

transmitted by the lower specimen.

2. Test specimens and miscellaneous equipment

The specimens used in the tests were hardened steel discs measuring two inches in diameter and 0.40 inches in thickness. A 16 millimeter hole at the center of each disc permitted attachment to the machine.

Before and between test runs the specimens were protected against rusting by storing them in a large glass desiccator which was charged with silica jel.

Loss of specimen weight was the manifestation of wear used in the tests. An analytical balance was used to weigh the specimens before and after each test interval. The specimens were cleaned in ethyl alcohol to remove oil and loose wear product before each weighing.

CHAPTER III

PROCEDURE

1. Preparation of specimens

All specimens were manufactured from a single length of $2\frac{1}{2}$ inch round, cold-rolled and annealed, tool steel. The material used was plain carbon steel of approximately 0.85 per cent carbon content. No analysis of the stock was obtained.

The specimens were machined 0.02 inches oversize by the machine shop prior to heat treatment. The center hole was drilled undersize inasmuch as it was learned from a pilot run that the inner diameter increased slightly during heat treatment.

The heat treatment was performed by the author and was similar to that used by Rosenberg [11] at the Bureau of Standards. The test pieces were given a uniform normalizing and hardening treatment. Eleven of the specimens were given various tempering treatments to obtain a range of hardness for one phase of the testing. The others received similar tempering treatments. A detailed account of the heat treatment is presented in Appendix II.

After heat treatment the specimens were returned to the machine shop for finish grinding and reaming. The sides were surface ground to 0.40 inches thickness, and the center hole was reamed to a diameter of 16 millimeters. Since the discs must fit snugly on the shafts of the machine, this dimension is a critical one, and a special reamer was procured for the operation. After reaming, the discs were mounted on a work arbor and wet ground to an outer diameter of 2.000 inches.

The hardness of the finished specimens was obtained using a Wilson Rockwell Hardness Tester. Several measurements were taken on a flat side of each disc as close to the periphery as was possible. Many of the test pieces showed considerable variation of hardness, and it was decided to perform tests on only those specimens with a reasonably uniform hardness. Hardness measurement data is tabulated in Appendix II.

The surface finish characteristics of nine representative specimens were examined using a Brush Model BL-103 Surface Analyzer [16]. Maximum roughness for the nine test pieces inspected varied from 275 to 375 microinches, and the average maximum roughness was 327 microinches. All specimens were similarly ground, and since those surfaces examined had maximum roughnesses of the same order of magnitude, no attempt was made to correlate wear and initial surface finish.

2. Conduct of the wear tests

Twenty-four runs were made, and at the beginning of Appendix III is a tabular presentation of the general scheme followed. Runs 1 thru 13 and 22 thru 24 investigated the effect of varying the load on specimens of uniform hardness. Runs 14 thru 21 were designed for determining the result of applying a standard 60 kg. load to upper specimens of varying hardness. The tests were patterned after a portion of the work of Rosenberg [11].

The procedure for all runs was essentially the same. The two specimens to be tested were washed in ethyl alcohol and permitted to dry. Each was then weighed on the analytical balance. The discs were mounted on the shafts of the Amsler machine using paper washers, and the retaining

nuts were tightened. At all times care was exercised to keep oil and other foreign matter from the wearing surfaces of the specimens.

The counter of the integrating mechanism was zeroed manually, and the proper weights and scale were fitted to the dynamometer. The swing was gently lowered until the upper specimen came to rest on the lower test disc, and the desired load was applied. Load in kilograms was indicated on a scale attached to the calibrated spring. The existing reading on the revolution counter was noted. The machine was started, and the test began.

When five thousand revolutions of the lower specimen had taken place, the machine was stopped. The integrator counter reading was recorded, and the test pieces were removed, washed in alcohol, permitted to dry, and weighed on the analytical balance. The discs were then replaced on the machine which was restarted and run for another five thousand revolutions.

A five thousand revolution test interval required about 25 minutes, and the manipulations between such periods of test took about six minutes. Except for Runs 5 and 6 the duration of the runs for the tests of uniform hardness was 50,000 revolutions. Runs 5 and 6 were at light load and continued for 135,000 and 200,000 revolutions respectively. In many runs the 35,000 and 45,000 revolution readings were omitted, and in a few cases the 40,000 revolution reading was also omitted.

Specimens of different hardnesses were placed on the upper shaft during eight runs. These specimens ran against lower discs of uniform hardness, and a standard load of 60 kg. was applied. In this series of runs each test proceeded without interruption for about 15,000 revolutions

until 100,000 meter-kilograms of frictional work was indicated by the counter of the integrating mechanism.

For Runs 1 thru 22 the diameter of the specimens was 2.000 inches. However, for the final two runs four previously tested specimens were reground to diameters of 1.985 inches and reused.

3. Use of the swing balancing arrangement

The weight of the swing which supports and drives the upper specimen is considerable. For runs at low loads the weight of the swing is balanced by a counterweight and pulley arrangement, and a smaller spring is used. This configuration of the machine was used for Runs 5, 6, and 22. Runs 5 and 6 were performed at indicated loads of 20 kg. and 10 kg. respectively. However, it was determined that the swing balancing arrangement did not equilibrate but instead overcompensated for the weight of the swing. Thus, the effective normal loads were much less for Runs 5 and 6 than those indicated by the scale of the calibrated spring.

For Run 22 at 20 kg. the counterweight and pulley arrangement was used, and additional weight was added to the top of the swing to bring it to the point of equilibrium. These three runs at low loads and a method for improving the swing balancing arrangement are discussed in the next section.

4. Calculation of effective loads

The swing was balanced for Run 22 by adding 4.6 kg. to the top of the swing. These additional weights were so positioned that a moment of approximately 28.5 kg.-in. was applied to the swing about the swing bearings. The line of contact for the two inch specimens was 5.17 in. from the swing bearings. Therefore, the additional normal load was $\frac{28.5 \text{ kg.-in.}}{5.17 \text{ in.}}$

or 5.5 kg. With the swing balanced Run 22 was performed at 20 kg. indicated and effective load. However, the swing was not balanced for Runs 5 and 6, and the indicated loads for those runs must be reduced by 5.5 kg. to obtain the effective loads. This reduction results in effective loads of 14.5 kg. and 4.5 kg. for Runs 5 and 6 respectively.

An additional weight of one kilogram was placed on the swing for the three runs at low loads. This running weight, as it is called by the manufacturer, compensates for the change in the position of the center of gravity of the swing when specimens 1.62 in. or greater in diameter are used. As placed during Runs 5 and 6, this weight applied a moment of 12.5 kg.-in. to the swing about the swing bearings. This weight was inadvertently left on the machine when the swing balancing accessories were removed from the swing after the completion of Run 6. It remained on the machine during Runs 7 thru 11, and thus the indicated loads for those runs must be increased by $\frac{12.5 \text{ kg.-in.}}{5.17 \text{ in.}}$ or about 2.4 kg. to obtain the effective loads.

The swing balancing arrangement should be modified before the machine is again used at low loads. The moment applied about the swing bearings by the counterweight is about 28.5 kg.-in. greater than it should be. By means of the pulley and strap the counterweight exerts on the swing a vertically upward force of 16.5 kg. The perpendicular distance from the swing bearings to the line of action of this force is 7.8 in. Thus, the amount of material to be removed from the counterweight should weigh $\frac{28.5 \text{ kg.-in.}}{7.8 \text{ in.}}$ or about 3.5 kg. This is equivalent to about 8 lbs., but it is recommended that material weighing slightly more than this be removed from the counterweight. Final balancing can

be done by adding material a bit at a time until the point of equilibrium is reached.

It was not understood why balancing of the counterweight system was not accomplished at the factory. Since it would be advantageous to have the machine balanced for the size specimens most usually employed, it is possible that the manufacturer wished to give an American purchaser the opportunity of balancing his own machine for the size specimens to be used as standard by him.

5. Data and data reduction

Data taken at each test interval included the revolution counter reading, the integrator counter reading, and the weights of the two test specimens. During the initial 5000 revolution test interval of Runs 1 thru 6 readings of the integrator counter and the torsional moment vs. revolutions were taken frequently without stopping the machine. Since the rate of work with respect to revolutions did not change abruptly during the initial interval of testing, these latter readings were discontinued after Run 6.

Data reduction was relatively simple and was accomplished as the testing proceeded. The revolution counter indicated total machine revolutions, and the number of test revolutions was obtained by subtraction. Total frictional work was calculated by multiplying the integrator counter reading by the value of the dynamometer scale as explained in Appendix I. Wear in terms of specimen weight lost was obtained by subtracting the weight found at the end of each test interval from the initial weight.

In a few cases it was neccessary to add weight to the dynamometer

pendulum during a test interval when the torsional moment increased to the limit of the scale. The readings of the integrator and revolution counters were noted at the time of the change, and an additional calculation was made to take into account this change of scale. Changing scales did not require stopping the machine.

For each of the runs using specimens of uniform hardness upper specimen weight loss, lower specimen weight loss, and frictional work were plotted against revolutions, and upper wear was plotted against frictional work. These curves and the test data for each run constitute Appendix III. These curves and data were used to prepare the curves of results contained in Chapter IV.

6. Calculation of the contact stresses

The maximum principle contact stress at the line of contact was calculated for the various normal loads. The method used [17] gave results in close agreement with Rosenberg [11]. The calculations and results are appended as Appendix IV.

However, this method is applicable to the case of cylinders in line contact with only a normal force acting. In the case of the test specimens the final coefficients of friction were about 0.3 or greater. With this large a tangential force acting, the principle stresses at the line of contact are very much greater than the maximum principle stress calculated for the case of normal force only. Tensile stresses as well as compressive stresses are present. In addition to this, the point of maximum shearing stress has moved from a few thousandths of an inch below the surface to the surface [17].

As a final complication to the problem, one must consider the

effect of the variable axial frictional force introduced by the reciprocation of the swing. The resulting problem is one with forces acting in three dimensions, and the solution was not attempted.

7. Additional comments

Only a single casualty occurred during the testing. At the end of Run 18 the lower specimen could not be removed from the shaft of the machine. Removal was finally accomplished using a small wheel puller and was not difficult in itself. However, the operation involved partial disassembly of the machine, and this took considerable time. No positive reason can be given for the cause of the casualty. Some heating of the specimens resulted from the testing, and perhaps in this instance the heating caused a relief of internal stresses remaining from the heat treatment. This stress relief could than have resulted in a slight change of dimensions.

Near the conclusion of the testing the roundness of the test pieces was checked with a dial indicator by a machinist during a search for the cause of a slight vertical reciprocation of the swing. Checks on six sample specimens showed that the specimens were 0.003 inches out-of-round. It was subsequently learned that the work arbor used in the final grinding of the specimens had not been a snug fit. No attempt was made to evaluate the effect of out-of-roundness of the specimens on the test results.

The manufacturer's instructions indicate the addition of a small braking element to the swing drive mechanism of the machine for cases when the axially reciprocating motion is used. This element was not delivered with the machine and was ordered from the manufacturer's

agent. However, it did not arrive from Switzerland until testing was nearly completed, and all tests were made without it. Without the braking element the angular velocity of the eccentric cam causing reciprocation of the swing is not uniform. No attenpt was made to evaluate the effect of this on the results.

CHAPTER IV

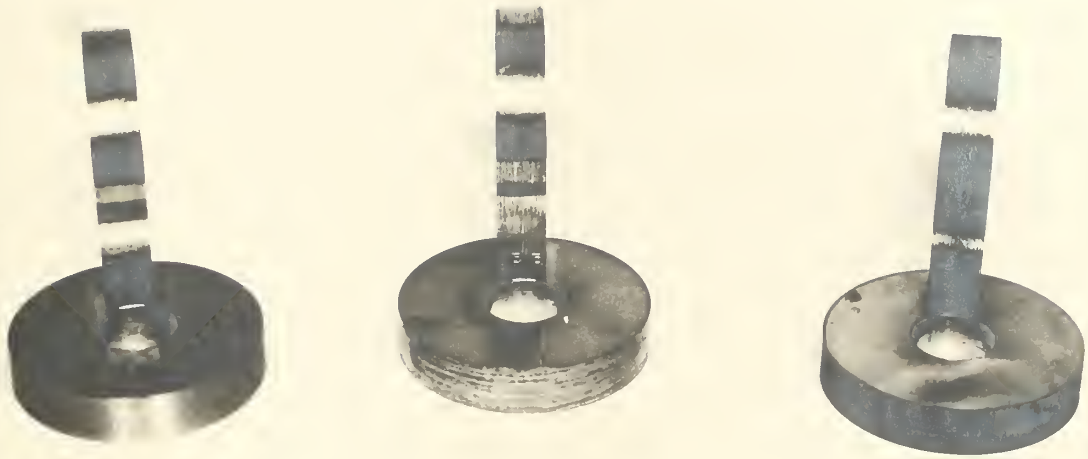
RESULTS

1. General observations

From an analysis of the test data it was determined that the wear of the steel tested varied greatly depending upon the specimen hardness and the normal load. However, from an observer's viewpoint the apparent behavior of each of the runs was quite similar.

In all cases extensive oxidation or scaling of the specimens occurred. The amount of scale or film formed was always greater on the lower, more rapidly turning test piece. As the run proceeded an upper specimen became smoother and shinier, and a lower specimen reached a fairly uniform reddish-black color. A photograph, Fig. 2 on the following page, enables a comparison of upper and lower specimens.

The general behavior of the frictional force, as measured by the dynamometer, was also similar during the 24 runs. At the start of a run the tangential frictional force quickly assumed a steady, low value. After a short and variable period of testing the frictional force began steadily and smoothly increasing in magnitude over a much longer but also variable test period. Upon attaining nearly the final average value the frictional force began oscillating, and these oscillations continued for the remainder of the test. As indicated by the smooth curves drawn of frictional work vs. revolutions, the average frictional force during the latter interval was constant. The period of the oscillations, as indicated by the rather violent swings of the dynamometer, was in the general magnitude of one second. No relationship



WEAR TEST SPECIMENS

LEFT: SPECIMENS BEFORE TESTING. NOTE THE APPEARANCE OF THE ORIGINAL GROUND SURFACE.

CENTER: UPPER SPECIMENS AFTER TESTING. THE FILM IS THIN AFTER 50,000 REVOLUTIONS.

RIGHT: LOWER SPECIMENS AFTER TESTING. THE FILM IS HEAVY AFTER 50,000 REVOLUTIONS.

LOAD 52.4 KILOGRAMS

Fig. 2. Appearance of the test specimens

was found between this period and that of the swing reciprocations.

Generally, the dynamometer was arranged for the 150 kg.-cm. scale before commencing a test run, and changes in the torsional moment in the initial few hundred revolutions were not readily apparent. However, for Run 6 the test was begun with the dynamometer set on the 10 kg.-cm. scale. Torsional moment for this low load run was observed to hold steady at 4.5 kg.-cm. for a few seconds after the swinging of the dynamometer due to starting the machine had damped. Torsional moment then dropped to a value of slightly over 4.0 kg.-cm., and there it remained for nearly four hundred revolutions before commencing the smooth and steady increase described above. It is believed that this small decrease occurred unobserved during the other runs and was indicative of the completion of the wearing in of the test pieces.

It is unfortunate that a detailed record was not kept of the counter readings when pronounced changes in the torsional moment and dynamometer behavior occurred. At the time of performing the tests the only data believed important were revolution counter reading, integrator counter reading, and specimen weights. While these three items are useful in evaluating the relative wear resistance of a material, they are by themselves of little use in a determination of just how the phenomenon of wear took place.

2. Frictional work and the coefficient of friction

Fig. 3 illustrates the variation of total frictional work with lower specimen revolutions during the initial thousand revolutions of Runs 1 thru 6. Fig. 4 and Fig. 5 provide the same information for 5000 revolutions and 50,000 revolutions of those six runs. The values of work

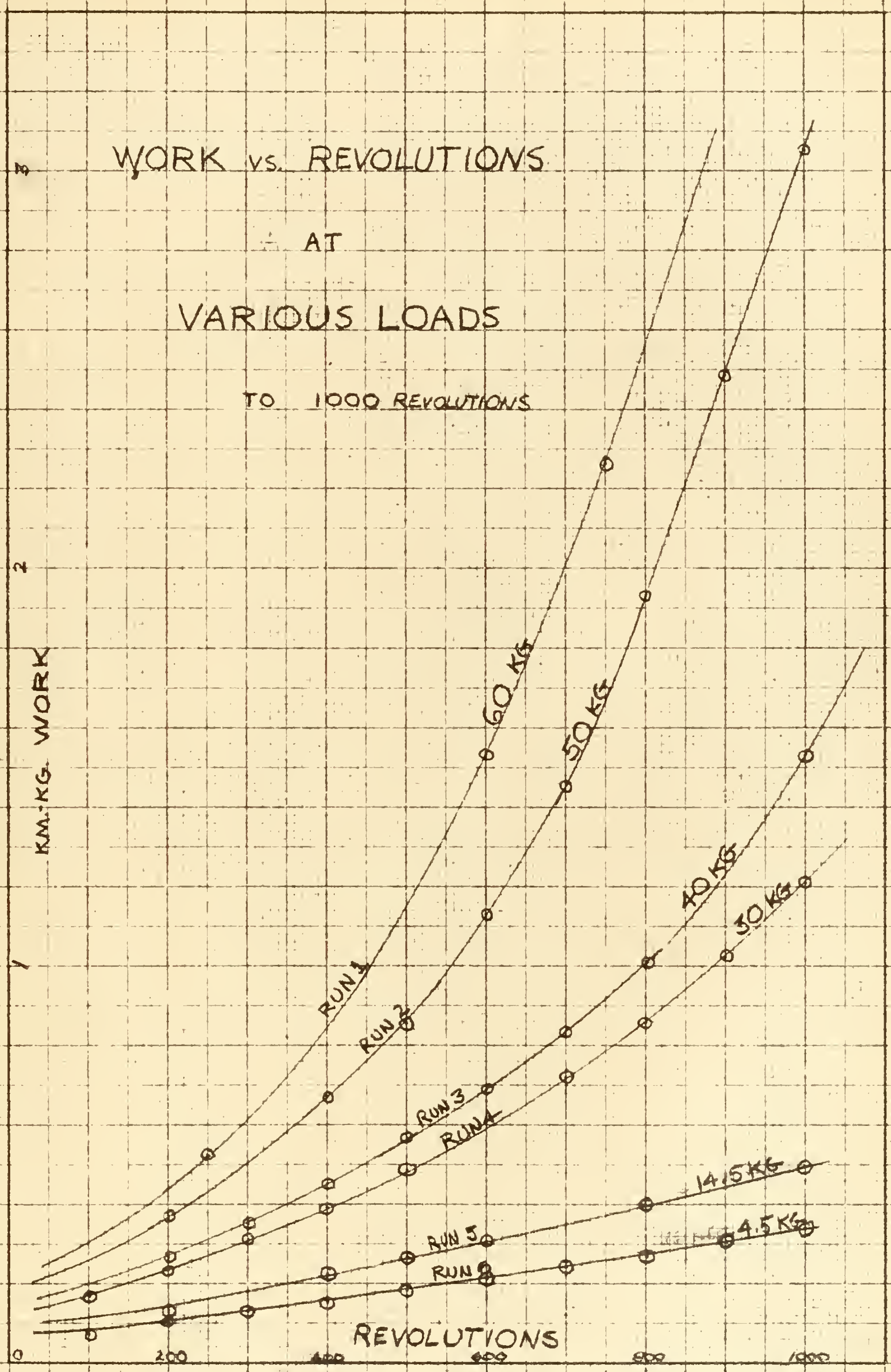


Fig. 3. Work vs. revolutions to 1,000 revolutions

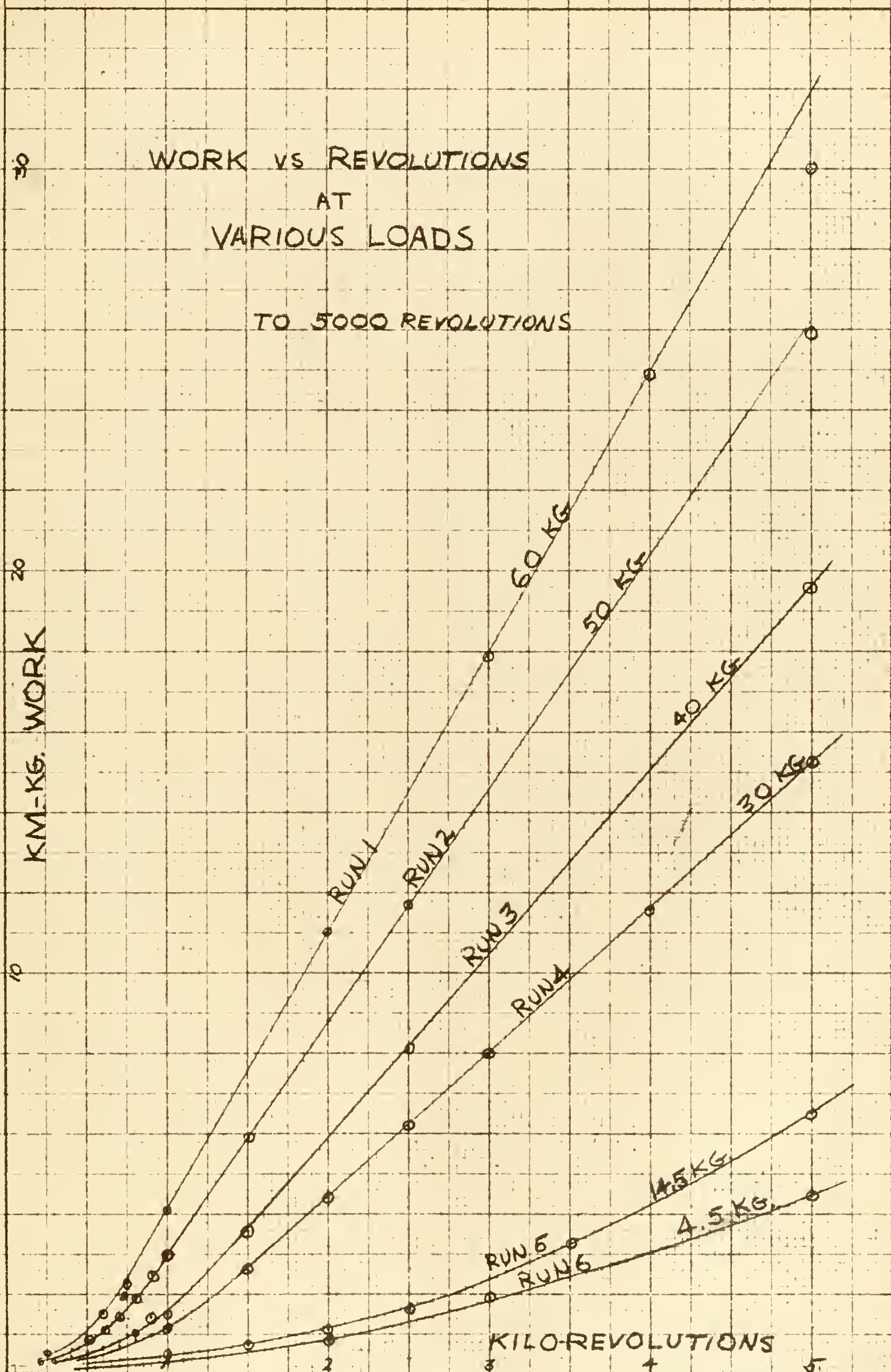


Fig. 4. Work vs. revolutions to 5,000 revolutions

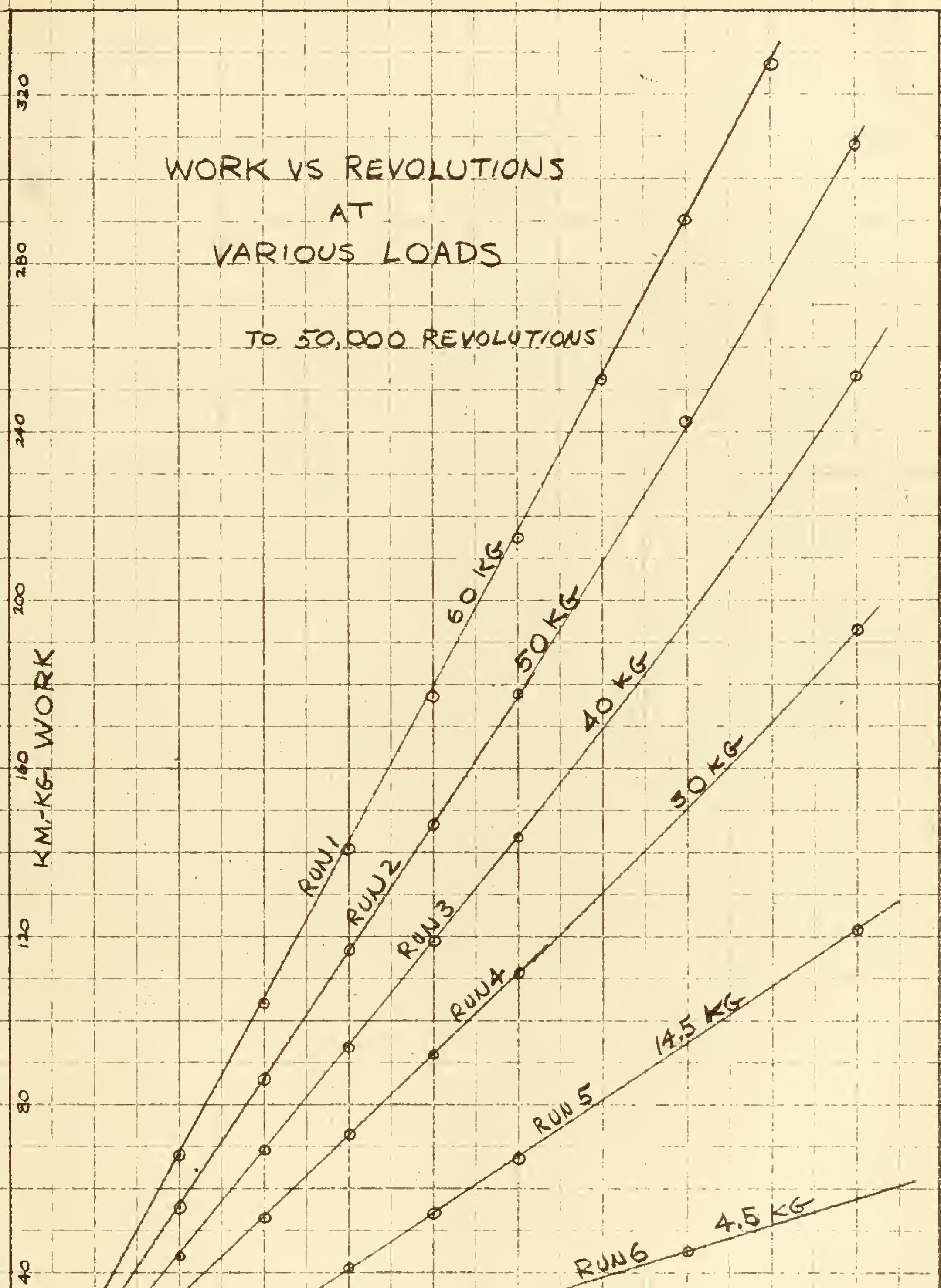


Fig. 5. Work vs. revolutions to 50,000 revolutions

were obtained from integrator counter readings at each test point. An explanation of the operation of the integrating mechanism is undertaken in Appendix I. Similar curves for each of the tests of discs of uniform hardness are presented in Appendix III with the test data.

Figs. 3, 4, and 5 are presented here to show the increase of the rates of work with respect to revolutions from initial small to final large and constant values. These slopes are proportional to the tangential frictional force. Since the ratio of the tangential frictional force to the normal force is the coefficient of friction, this phase of the test results was considered an important indication of the changes taking place during the wear process.

It will be observed that the slope of each curve increases to a constant, final value. These slopes are the rates of work with respect to revolutions or distance traveled, and a sample calculation will illustrate the procedure followed in obtaining the final coefficients of friction for the 16 runs.

In Fig. 5 the value of the final slope for Run 1 is 7.45 kilometer-kilograms per thousand revolutions or 745 centimeter-kilograms per revolution. The tangential frictional force is applied thru a distance of $2\pi \times 2.54$ cm. per revolution for the case of two inch diameter test pieces. Therefore, the average frictional force corresponding to the slope under consideration is $\frac{745 \text{ cm.-kg.}}{2 \times 2.54 \text{ cm.}}$ or 46.7 kilograms. The normal load for Run 1 was 60 kg. Thus, the final coefficient for Run 1 was 0.778. Similar calculations were made for Runs 2 thru 13 and for Runs 22 thru 24, and the results appear in Fig. 6.

From the initial torsional moments observed for Runs 1 thru 6 the

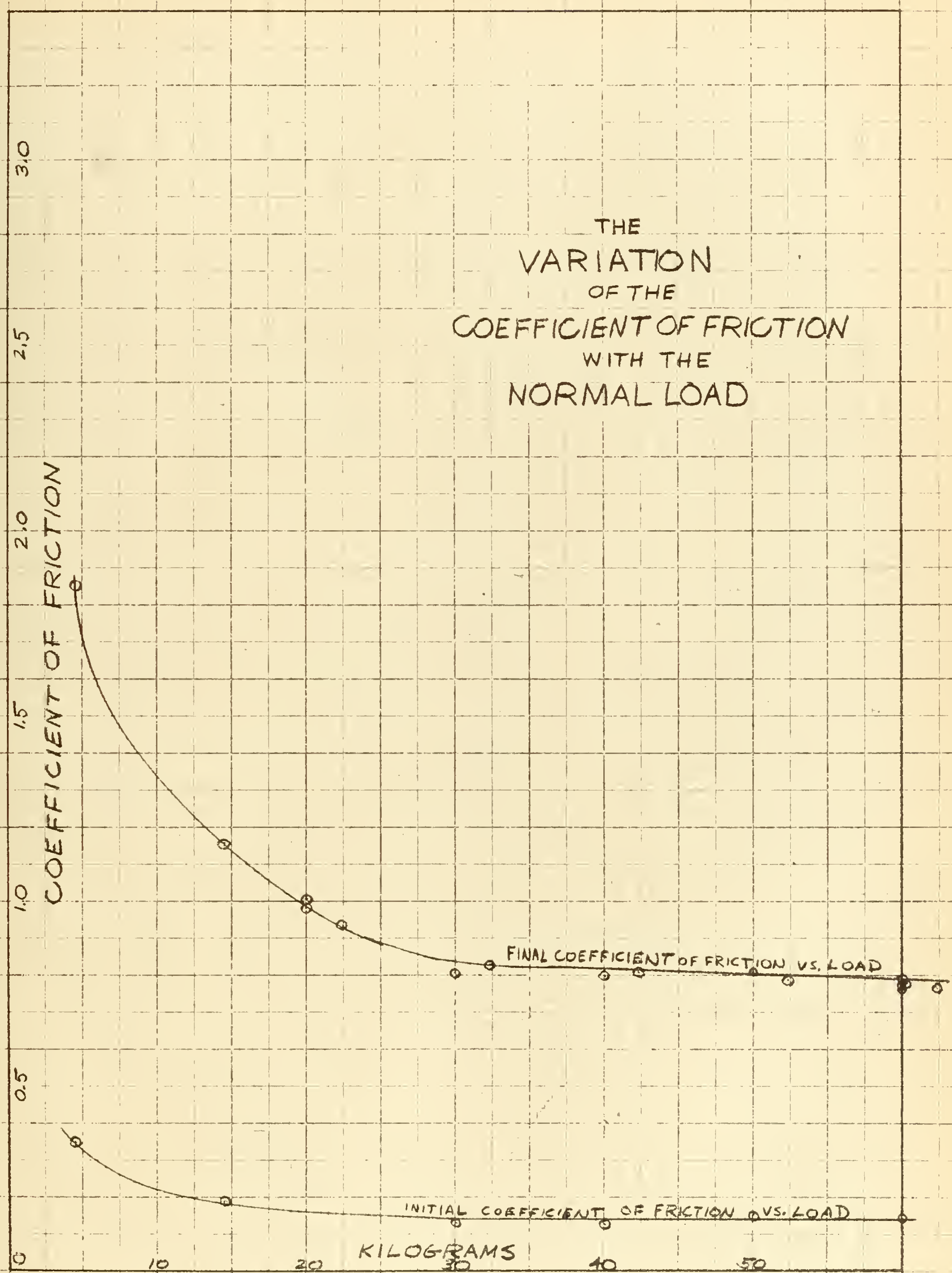


Fig. 6. The coefficient of friction vs. normal load

initial coefficients of friction were computed. Presumedly, this coefficient of friction pertains to the short time interval between the rather immediate wearing in process and the increase in oxidation of the test pieces. The initial torsional moment in kilogram-centimeters divided by the radius of the lower specimen in centimeters gave the frictional force in kilograms. The initial coefficients of friction are also plotted against normal load in Fig. 6.

Fig. 6 illustrates a definite deviation from Amontons's second law [6]. This so-called law states that the coefficient of friction is independent of the load. It holds for a wide variety of experimental conditions involving clean surfaces and for surfaces lubricated under boundary conditions. However, as Bowden and Tabor [7] point out, the law holds only when an increase in load causes a proportionate increase in the actual or intimate area of contact. An oxide film on the surfaces would presumably prevent this increase. Fig. 6 shows that the greater deviation from Amontons's law occurs in the case of the final coefficients of friction when the scale is quite heavy. For the curve of initial coefficients of friction vs. load the deviation is less pronounced. At that time no increase in oxidation was apparent, but the specimens were not clean in the laboratory sense of the word. Evidently, the adsorbed oxides were enough to cause the slight deviation from Amontons's law observed at low loads.

3. The effect of work on the wear

For the 16 runs conducted with specimens of uniform hardness it was found that the wear of the upper specimen was linearly related to the frictional work. The rates of upper specimen wear with respect to work

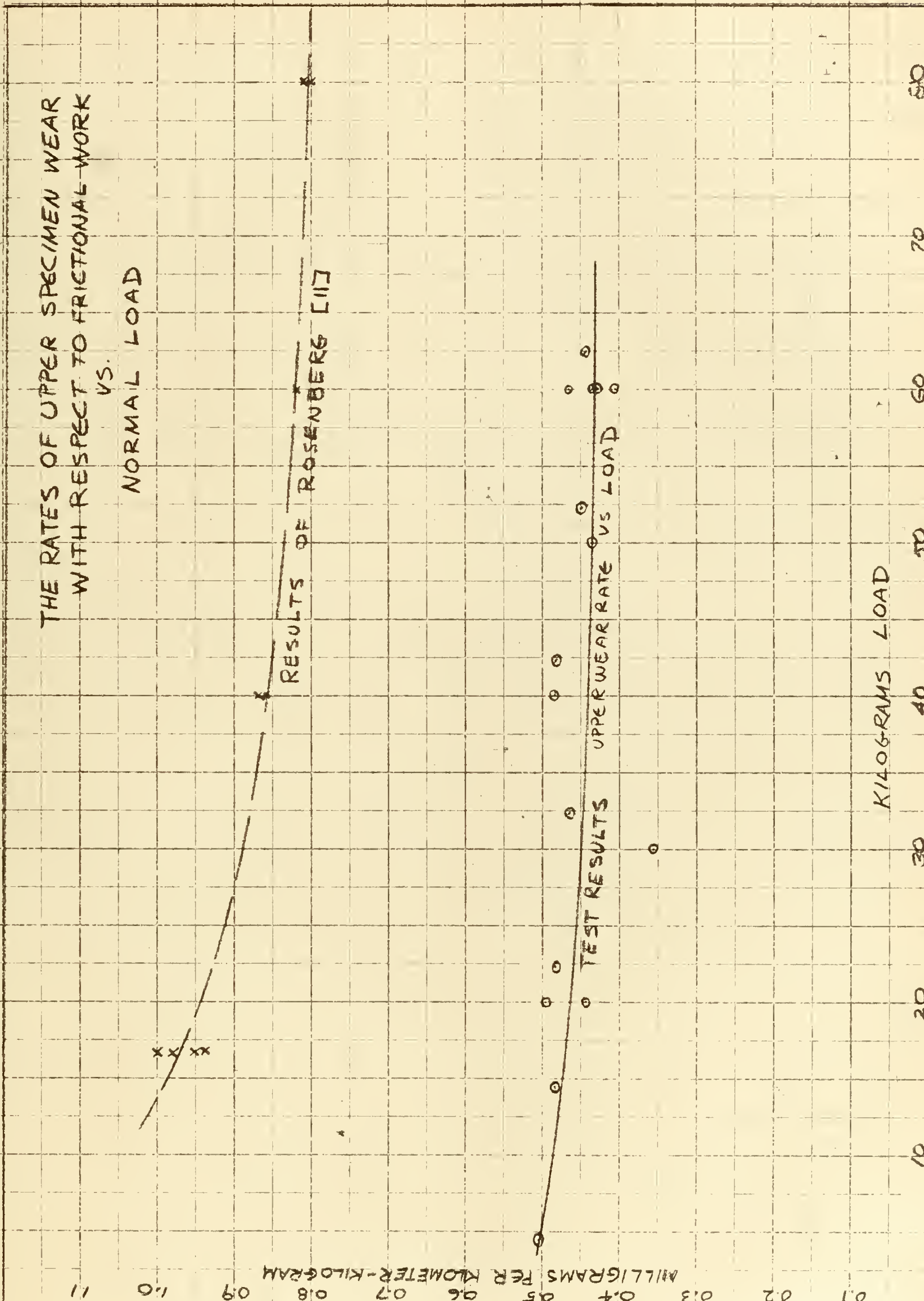


Fig. 7. Rate of upper wear with respect to work vs. normal load

were determined and are plotted against the normal load in Fig. 7. The rates at first appeared to vary randomly within fairly narrow limits, but a smooth curve was drawn to show a decreasing wear rate with increasing load. This is in agreement with the results of Rosenberg [11], although in his curve the decreasing wear rate with increasing load is much more evident. For the sake of comparison, Rosenberg's results are also plotted in Fig. 7. His tests were conducted using an Amsler machine and two inch discs, and the results plotted are for tests performed on plain carbon steel of 0.81 per cent carbon. The present author's tests were largely patterned after those of Rosenberg's, and the specimens were given a smilar heat treatment. Yet, though the material was supposedly of nearly the same composition, a striking difference in the rates of upper wear with respect to work is apparent.

Rosenberg reported that the rate of lower specimen wear with respect to work increased with load. In the 16 runs performed with specimens of uniform hardness by the present author it was found that the relationship between lower specimen wear and work was rarely linear. Therefore, no attempt has been made to plot lower wear rates on Fig. 7.

Figs. 8 and 9 show the relationship of work and upper and lower specimen wear. The figures were obtained by plotting all of the wear data against the corresponding data for frictional work.

4. The effect of hardness on upper specimen wear

The hardness of the upper specimen had a pronounced effect on the wear of the upper specimen, and this effect is illustrated graphically in Fig. 10. Upper specimen weight loss during the initial 100 kilometer-kilograms of work is plotted against upper specimen hardness. Runs 1,

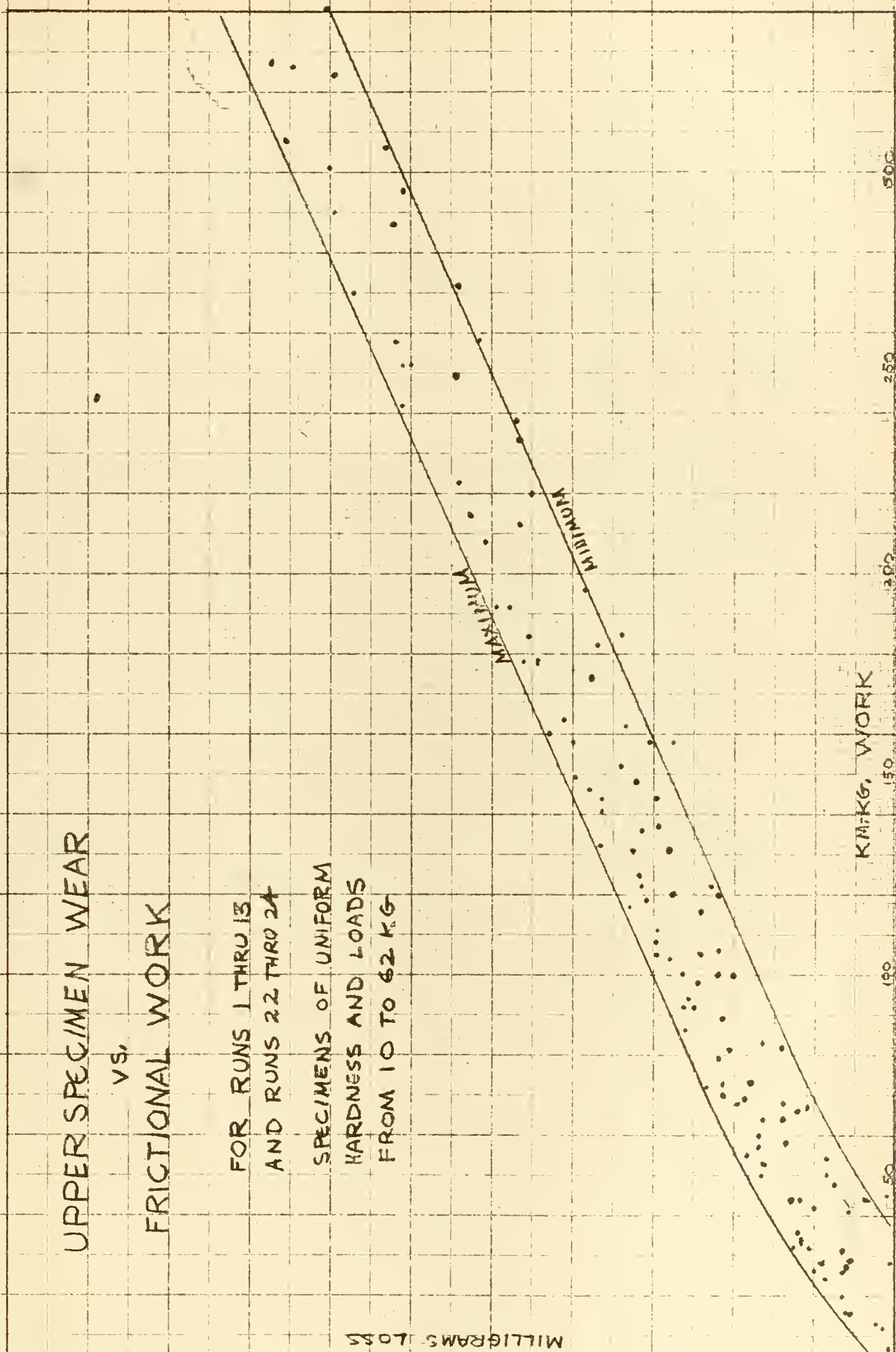


Fig. 8. Upper wear vs. frictional work

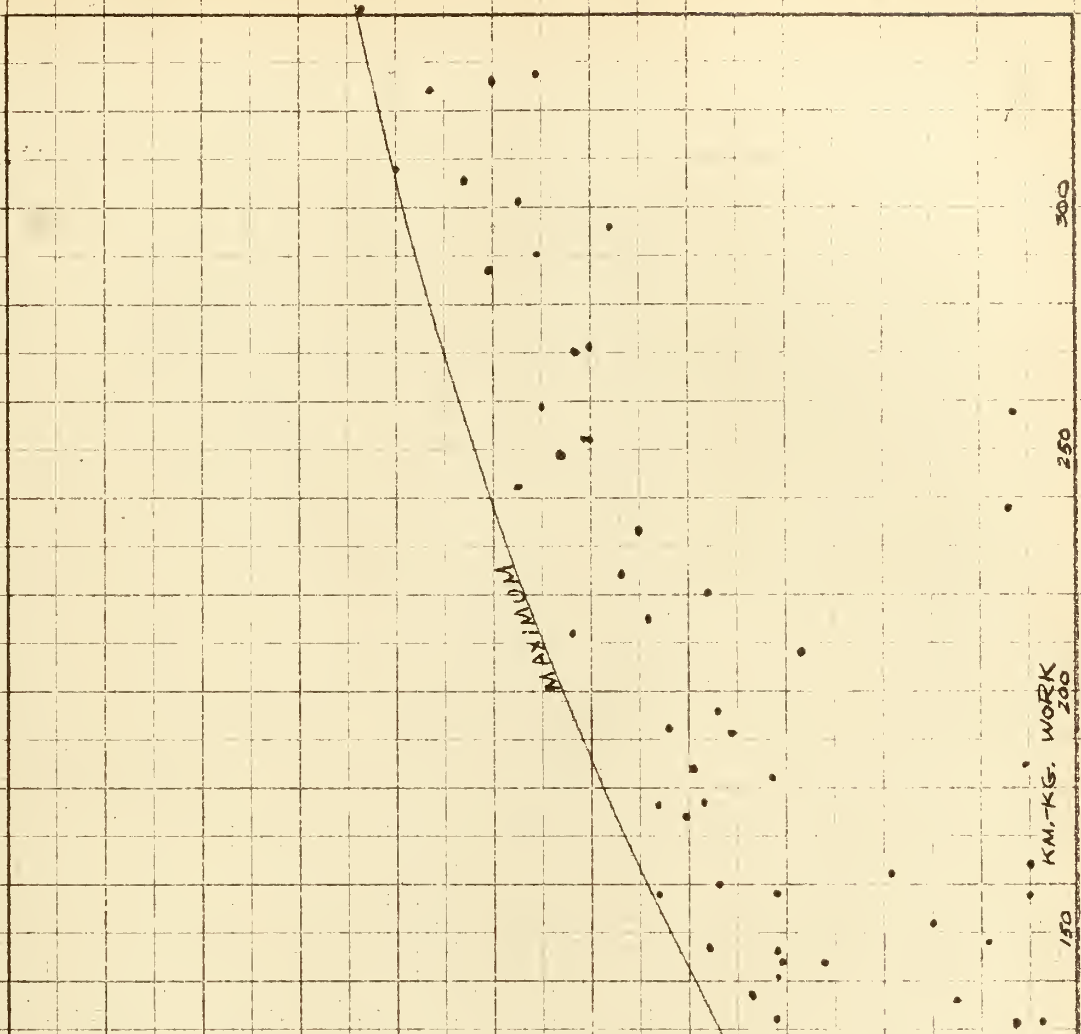


Fig. 9. Lower wear vs. frictional work

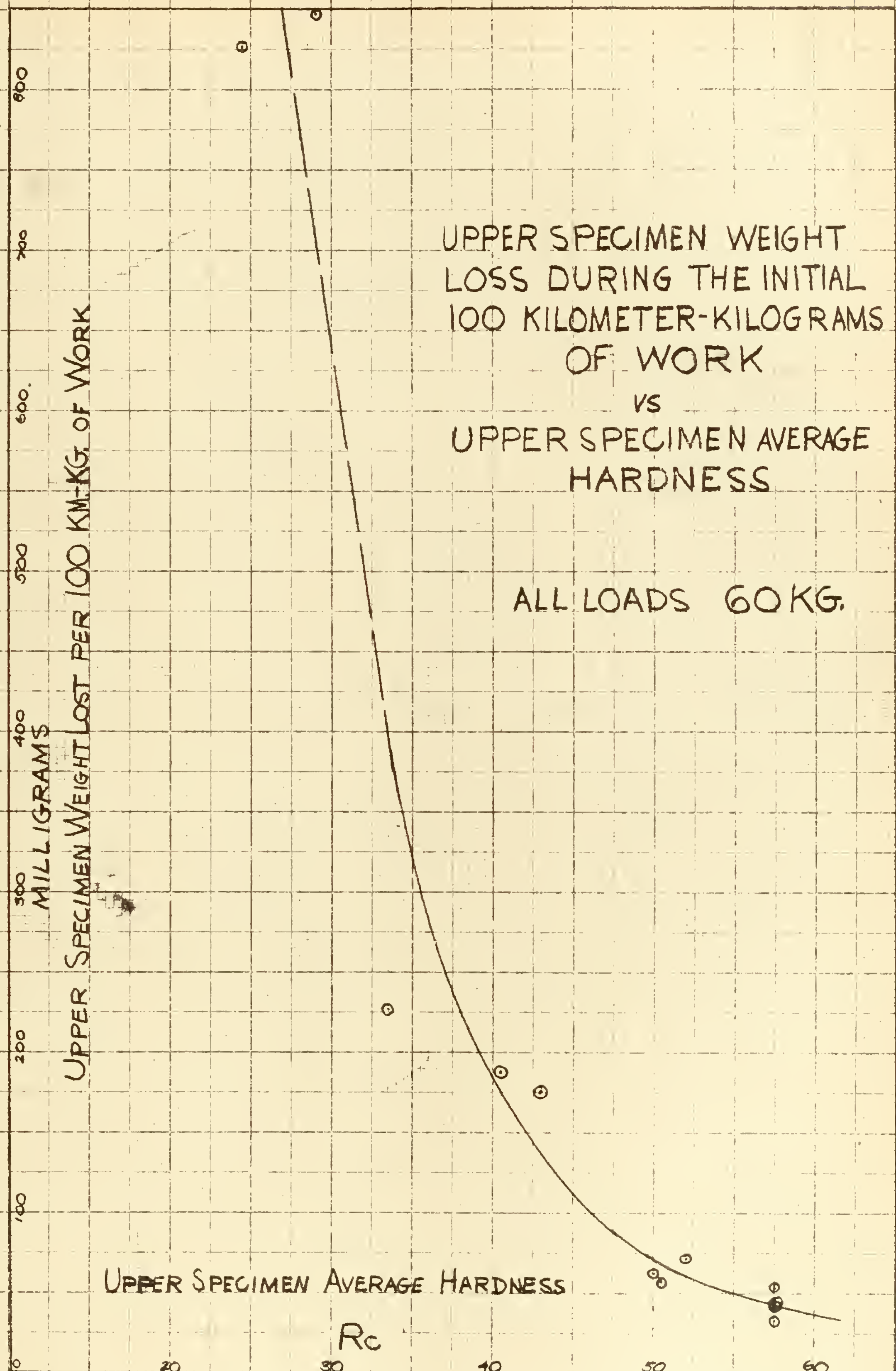


Fig. 10. Upper wear vs. upper hardness

13 thru 21, 23 and 24 are reflected on the curve. The load was 60 kg. in each instance, and lower specimens of uniform hardness were used. It will be observed that the wear resistance of the hardest specimens was nearly twenty times greater than the wear resistance of the softest specimens.

It may be of interest to note that the number of revolutions required to obtain 100 kilometer-kilograms of work at 60 kg. load was very close to 15,000 regardless of the amount of wear occurring. Since the work is the result of the frictional force operating over a distance, it can be concluded that the coefficient of friction is independent of the magnitude of the wear involved under the test conditions at least.

5. The effect of the distance traveled on the wear

Fig. 11 consists of eight curves for representative loads, and the linear relationship between upper wear and revolutions or distance traveled is shown. Obviously, a correlation exists among this linear relationship, the linear work vs. revolutions curves, and the linear upper wear vs. work curves in Appendix III.

The rates of upper wear with respect to distance traveled were calculated for the 16 runs at uniform hardness, and these rates are plotted against the normal load in Fig. 12. The relationship between this rate and the normal load appears to be linear over the range of the test data.

Lower specimen wear also increased with increasing load, but as stated earlier, the lower wear vs. revolutions curve was rarely linear for very long. Therefore, no effort was made to derive the lower wear rates with respect to distance traveled.

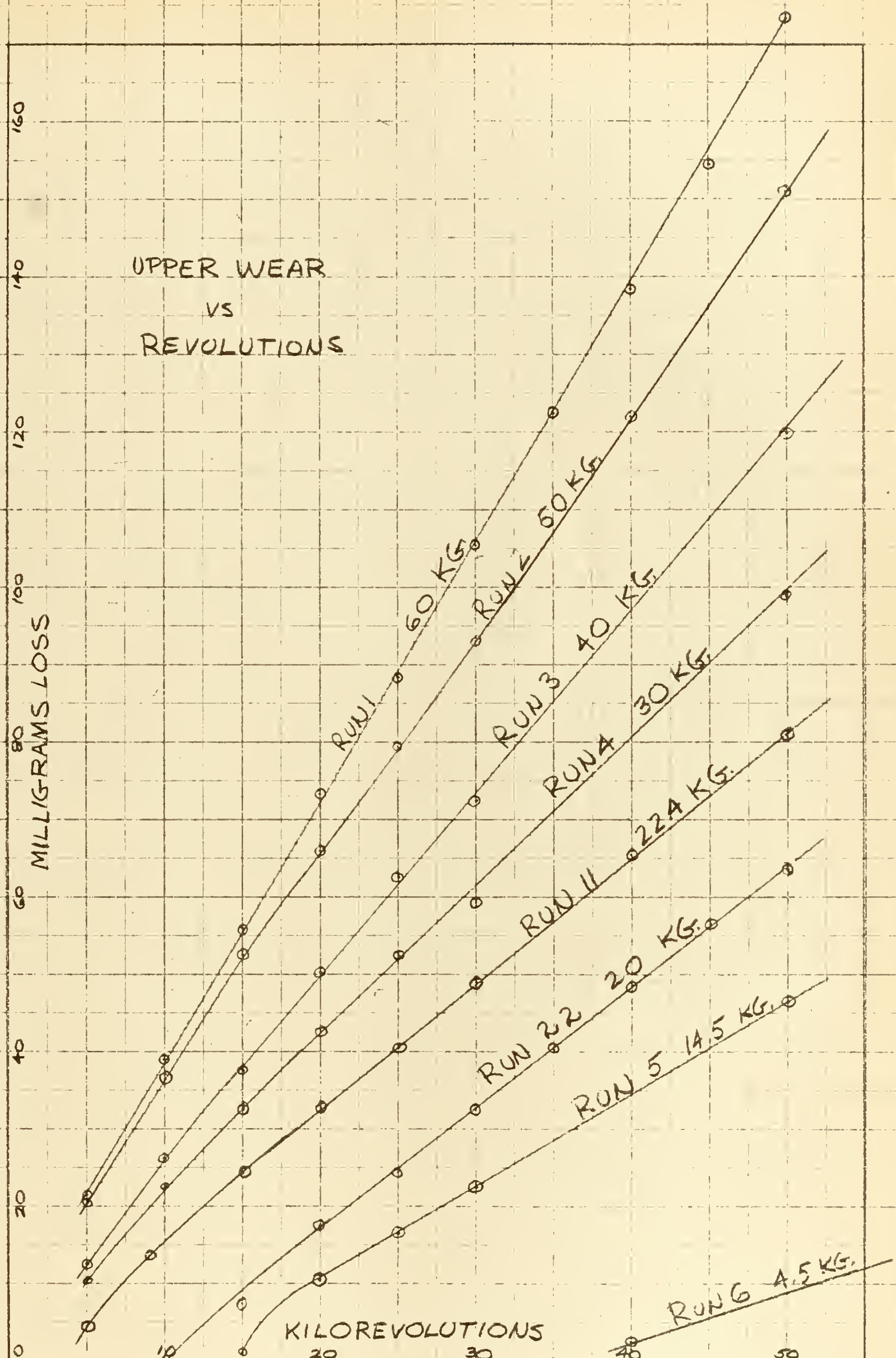


Fig. 11. Upper wear vs. distance traveled

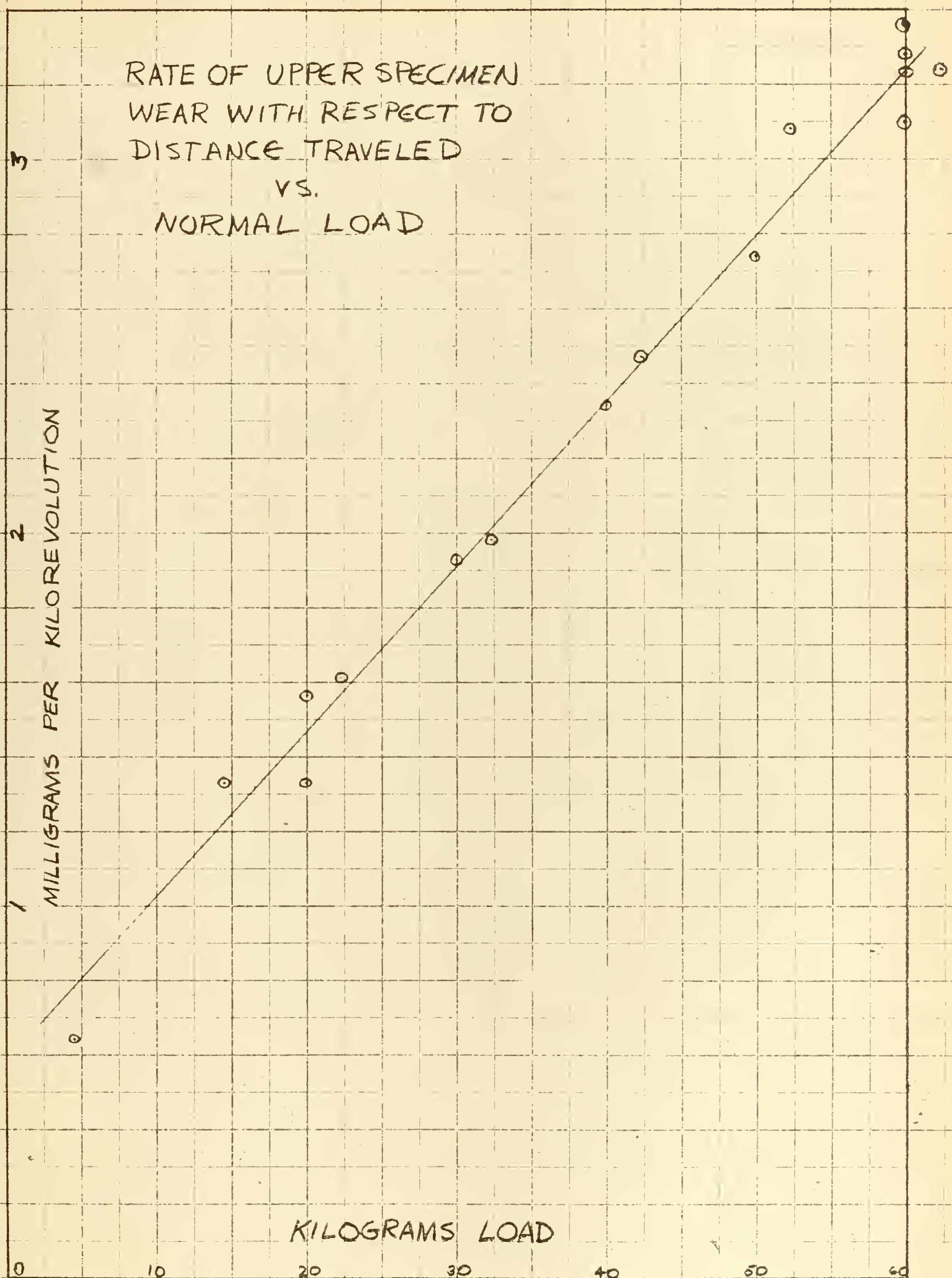


Fig. 12. Rate of upper wear with respect to distance traveled vs. normal load

6. Duplication of test results

Fig. 13 offers for comparison the results of wear vs. revolutions for Runs 1 and 13. Both runs were performed on specimens of nearly the same average hardness and were conducted using a normal load of 60 kg. As is readily evident, the duplication of results was not close. Close agreement was obtained for the curves of work vs. revolutions, but these curves were not drawn to avoid confusion. No effort was made to control the relative humidity of the test room, and since relative humidity would perhaps influence the degree of oxidation, this may be the reason for the disagreement.

In any repetition of a run made one may expect that the coefficient of friction, the rate of upper wear with respect to work, the rate of upper wear with respect to distance traveled, and the work vs revolutions curve will be closely the same as before. However, due to relative humidity and other variables it is likely that the other curves will not be coincident with the earlier curves.

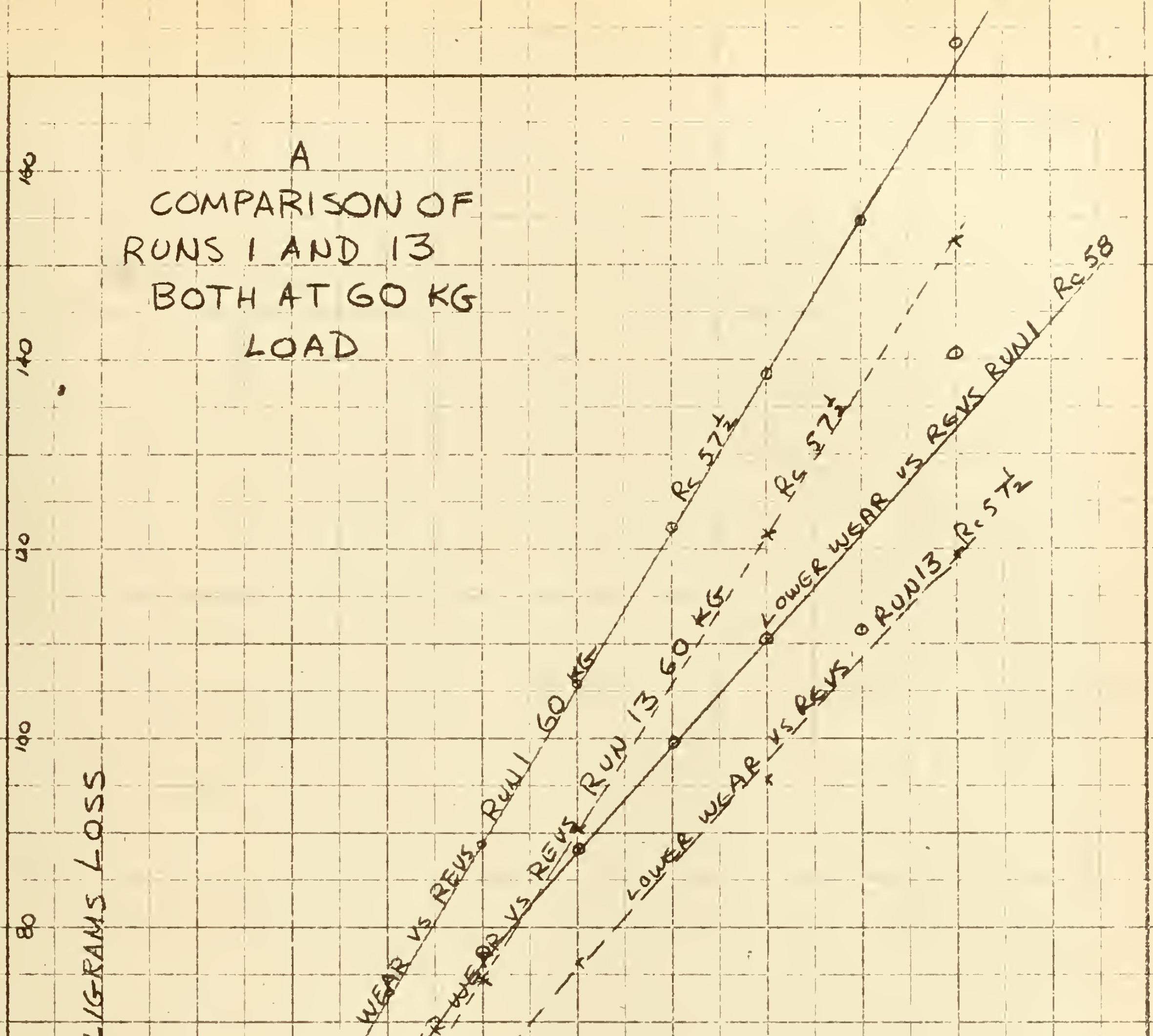


Fig. 13. A comparison of similar runs at 60 kg.

CHAPTER V

CONCLUSIONS

From the results of the 24 test runs the following conclusions are drawn. The conclusions are applicable to the material tested under the conditions of the tests. No conclusions may be drawn for this material under other conditions of operation.

The rate of work with respect to distance traveled was essentially independent of the upper hardness. The coefficient of friction was essentially independent of the hardness of the upper specimen. The rate of work with respect to distance traveled and the coefficient of friction were independent of the apparent area of contact during the early part of each run. During this phase of each run these items increased smoothly without observed fluctuations from relatively low initial values to relatively high final average values. These final average values of the rate of work with respect to distance traveled and the coefficient of friction were independent of the distance traveled, and the interval where this constancy prevailed constituted the bulk of each test run. During this period the instantaneous values of the rate of work with respect to the distance traveled and the coefficient of friction were not constant but oscillated considerably. The frequency of oscillations was not measured but was essentially constant during any single run. The period of the oscillations was of the magnitude of one second. Neither the initial nor the average final coefficients of friction obeyed Amontons's second law. Both coefficients of friction increased sharply at low loads.

The rate of upper wear with respect to work increased sharply with decreasing hardness. This rate was nearly independent of the normal load, decreasing only slightly as the load increased.

The rate of upper wear with respect to the distance traveled was directly proportional to the normal load and inversely proportional to the hardness of the upper specimen. This rate was essentially independent of the coefficient of friction.

The coefficient of friction, the rate of upper wear with respect to work, the rate of upper wear with respect to the distance traveled, and the work vs. revolutions curve were essentially the same for different runs performed under similar conditions of loading and hardness. However, duplication was not obtained for curves of upper and lower wear vs. revolutions.

The relationship between lower wear and revolutions was rarely uniform. Often the weight increased slightly during the first five or ten thousand revolutions. Generally, the lower specimen lost less weight during any test interval than the upper specimen. In all cases the film of oxides was greater on the lower specimen than on the upper specimen.

The swing balancing arrangement of the testing machine did not equilibrate the weight of the swing for low load tests. With this lone exception it can be stated that the machine operated satisfactorily. The integrating mechanism gave excellent results, and the initial zeroing of the dynamometer and integrator was readily accomplished.

CHAPTER VI

RECOMMENDATIONS

1. Modification of the testing machine

It is recommended that the counterweight system be modified as outlined in Chapter III before the machine is used again for tests at low loads. Unless it is foreseen that the machine will be used to test specimens a good deal smaller than two inches in diameter, it is recommended that the counterweight system be balanced for two inch specimens to facilitate future testing. Such balancing would not preclude the testing of smaller specimens, for in that event weight could be added to the cavity in the counterweight.

Installation of a limit switch would permit continuous, long-time testing to proceed with the machine left unattended. As explained in Chapter IV, an oscillating torsional moment occurs after the testing has reached a certain phase. The swings of the dynamometer pendulum which result from this oscillation become quite violent when high frictional forces are present. Occasionally the weights of the pendulum will strike the underside of the machine bed, and an operator should be in attendance to prevent damage to the machine. A suitable switch fastened to the point of dynamometer contact at the underside of the machine bed could open the power supply and thus turn off the machine. With such an accessory an operator would be required only for making adjustments to the machine and for recording data.

The revolution counter of the machine was damaged during shipment from the factory. It performs its designed function well, but it is

likely that operation in its present condition will lessen the expected service life. A new unit should be ordered from the manufacturer's agent, and it will then be available when needed.

2. Use of the Surface Analyzer

The Brush Surface Analyzer with its recorder and amplifier provides a magnified profile of the surface characteristics analyzed. An accessory, the averaging meter, was received without the proper connectors. This latter unit gives the average roughness or root-mean-squared value of roughness as desired for the surface measured. It is believed that either of these roughness readings could be useful in a study of the effect of surface finish on the wear resistance of a metal. Without the averaging meter the value of the equipment in connection with wear testing is questionable, for it is believed that the effect of any few sharp peaks affect the torsional moment of the machine for only a few revolutions. Yet it is the sharp peaks which give the maximum roughness reading.

It is recommended that no study of the effect of initial surface finish on wear be undertaken without including specimens which have been superfinished. This relatively new method of preparing smooth surfaces is claimed by its proponents to have a paramount influence in increasing the wear resistance of internal combustion engines and other applications where lubrication is a problem.

3. Proposed tests

It would be most interesting to investigate the effect of various liquids on the rate of wear with respect to work. It is believed that the rates of upper wear with respect to the frictional work would be

very similar regardless of the medium used. Obviously, many more revolutions would be required to perform a given unit of work, but it is believed that the wear rates would be the same for a given material with a given hardness.

Duplication of the apparatus and tests of Fink [12] presents a challenge. In Amsler tests of steel at one per cent slip and 50 kg. normal load, Fink obtained considerable wear and oxidation over a test interval. He then performed the same test in a nitrogen atmosphere and reported 0.0000 grams loss. American investigators [13] have attempted to duplicate his results, but it is not believed that any have succeeded. Tests in an inert atmosphere might also be attempted.

It is recommended that for most testing greater attention be devoted to the behavior during the first few thousand revolutions. Long-time tests can determine the relative resistance to wear of a number of metals, but once extensive oxidation has commenced, it is believed that the wear process is closely related to grinding, and little of fundamental importance is apt to be discovered. In any event appropriate counter readings should be taken when the torsional moment commences to oscillate. This probably has some simple relationship to the oxidation process.

One final suitable project that comes to mind would be an investigation of the wear resistance of teflon covered steel test pieces. Bowden and Tabor [7] report that the frictional properties of teflon are very good, and compare it to ice on ice. The coefficient of friction is reported as 0.04 without lubrication.

4. Procurement of specimens

It was learned that obtaining a specific specimen material was quite

difficult, and that where a thesis project is involved, a second best material is better than none. It is believed that for many investigations of the wear and friction phenomena the specific material is not too important. An analysis of the material should be requested, however. Since hardness is of such great importance, no effort should be spared to obtain uniform hardening. In the author's tests uniform hardness was a relative term only. Because of non-uniform heat treatment or poor procedures considerable variation existed among the specimens, and in some cases an individual wearing periphery had a non-uniform hardness.

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APPENDIX I

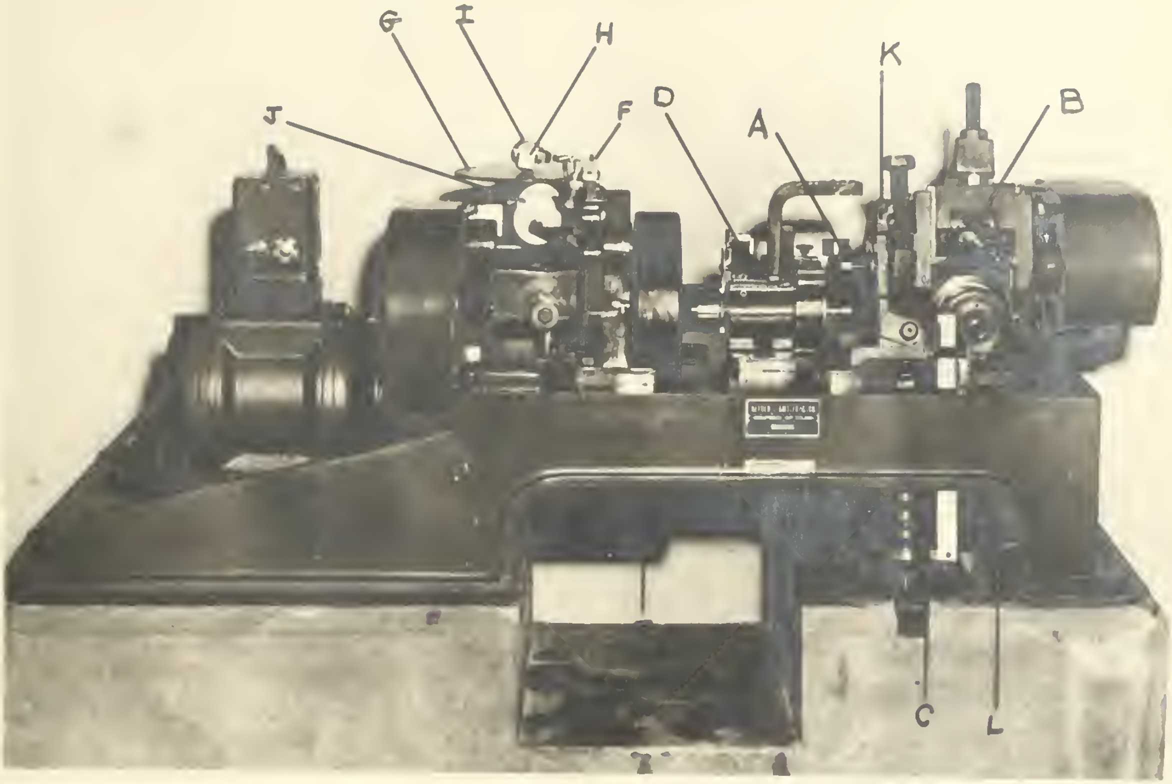
THE AMSLER WEAR TESTING MACHINE

Only two of the many possible configurations of the Amsler machine were used for the author's tests. The purpose of this appendix is to describe other arrangements of the machine [14] and to discuss certain features of the machine in greater detail than was done in the text. Fig. 14, which appears on the next page, is a photograph of the testing machine.

In the author's tests the machine was arranged to provide a combination of rolling and sliding friction. The reciprocating axial motion of the upper specimen was used at all times. Although different loads were applied and while different dynamometer scales were used, the only major change imposed was the use of the swing balancing arrangement and weak spring for three of the runs.

In addition to the axially reciprocating motion of the swing and upper specimen, the machine can impose a vertically reciprocating motion upon the swing and upper specimen. In this case the load is periodically applied and removed, and tests under conditions of impact loading can be conducted. The vertical motion of the swing can be obtained without the horizontal motion. Naturally, tests can be conducted with neither of these periodic motions if desired.

One can select either of two possible frequencies for this periodic motion. Since the manufacturer recommends the use of low speed of the swing for tests where the swing balancing arrangement is to be used, this speed was used in the tests. In addition to the two swing drive speeds



THE AMSLER WEAR TESTING MACHINE

A	TEST SPECIMENS
B	SWING
C	LARGE SPRING AND LOAD SCALE
D	REVOLUTION COUNTER
E	DYNAMOMETER WEIGHTS
F	DYNAMOMETER SCALE
G	INTEGRATOR PLATE
H	INTEGRATOR COUNTER
I	INTEGRATOR ROLLER
J	DIAGRAM RECORDER
K	LOAD ADJUSTMENT
L	COUNTER WEIGHT

(THE SWING BALANCING ARRANGEMENT IS NOT SHOWN.)

Fig. 14. Features of the Amsler machine

mentioned above, the operator can choose either of two speeds for the electric drive motor. Low speed, which was the speed used in the tests, is about 200 r.p.m., and high speed is twice the lower speed. Operating the machine at high speed would double the frequency of the swing motion as well as the speed of the test pieces.

By shifting gears the rotation of the upper shaft can be reversed. Greatly increased slip is then obtained, for the specimens are rotating in the same direction. Another gear change permits tests of sliding friction alone. The upper shaft is locked, and the upper specimen could be made in the form of a slider or of a shoe rather than a disc. Still another arrangement of the gears permits the upper shaft to rotate freely, and the wear under conditions of pure rolling friction could be tested. All gear changes are speedily accomplished. The only tool needed is a screwdriver, and the time required to make a change is about a minute.

The specimens can be run without lubrication as in the tests described herein. Accessories can be installed to add a metered amount of abrasive to the line of specimen contact. A reciprocating linkage stirs the abrasive and aids uniform flow, Other means are provided for supplying the test pieces with oil or other liquid and for collecting the excess lubricant.

The instantaneous torsional moment required to turn the lower specimen is the product of tangential friction and lower specimen radius. This moment is measured by a dynamometer. By changing weights and scales the operator can measure moments in the 0-10 kg.-cm., 0-50 kg.-cm., 0-100 kg.-cm., and 0-150 kg.-cm. ranges. The 150 kg.-cm. value of torsional moment is a top limit for the operation of the machine, for beyond that point

dynamometer travel is prevented by the underside of the machine bed. Greater moments can not be measured, and moments greatly in excess of 150 kg.-cm. would surely damage the machine.

The radius of the lower test piece is a constant for a given test. Therefore, it is really the tangential frictional force which limits the severity of the test. This is a function of the coefficient of friction as well as the applied load. Thus, the maximum load which can be applied during a given test is limited by the conditions of the test. The spring is calibrated to 200 kg., but it seems likely that this high a load could be applied only to lubricated specimens where the coefficient of friction would be small. Rosenberg [11] used a top load of 80 kg. in his tests of unlubricated steel discs. The present author attempted to duplicate this part of Rosenberg's work, but it was found that the maximum load which could be applied without reaching the limit of the machine used was about 65 kg.

The total frictional work transmitted by the lower specimen is given by a Veeder counter which is actuated by the roller of an integrating mechanism. The roller rests on a horizontal, circular plate which is geared to turn 1.62 times for one hundred revolutions of the lower specimen. The axes of the roller and plate are at right angles. The roller is linked to the dynamometer in such a way that its position varies from the center of the plate to a point 80 mm. from the center depending on the torsional moment. When there is no tangential friction, there is no torsional moment, and the roller will be at the center of the plate where it does not turn. The lack of motion of the counter indicates that no work is being performed. However, at full dynamometer deflection the roller is displaced

80 mm. from the center where it turns rapidly and gives the maximum indication of work.

The diameter of the roller is 41.3 mm., and thus, for one revolution of the plate the roller turns $\frac{2 \times 80}{41.3} = 3.874$ times in the case of full dynamometer deflection. For one hundred lower specimen revolutions the plate turns 1.62 times, and in the case of full deflection of the dynamometer the roller will then turn $1.62 \times 3.874 = 6.28 = 2\pi$ times. For lesser values of torsional moment, m, the roller will turn $2\pi \times \frac{m}{M}$ times for each hundred revolutions of the lower test piece. Here M is the torsional moment at full dynamometer deflection and would be 10, 50, 100, or 150 kg.-cm. depending the scale used.

The work transmitted by the lower specimen in one revolution is the product of distance traveled, $2\pi r$, and the tangential frictional force, F. Where r, the lower specimen radius, is measured in centimeters and F is measured in kilograms, the work per hundred revolutions is $100 \times 2\pi r \times F$ cm.-kg.

Dividing work, $100 \times 2\pi r \times F$ cm.-kg. per hundred revolutions of the lower specimen, by the roller indication, $2\pi \times \frac{m}{M}$ turns per hundred lower specimen revolutions, one obtains the integrator constant, $\frac{100 \times r \times F \times M}{m}$ cm.-kg. per roller turn.

However, $m = r \times F$. Therefore, the integrator constant is 100 M cm.-kg. per turn. A more convenient unit of work is the meter-kilogram, and to obtain work in m.-kg., one need only multiply the value of the integrator counter by M, the dynamometer scale.

The roller of the integrator can be relocated on the machine to give an indication of specimen wear. In this case the roller bears on the

lower specimen which is turned one hundred times with the upper specimen disengaged. Since the diameter of the roller is 41.3 mm., observation of the change in counter reading will enable calculation of the specimen diameter. In this case reduction of diameter would be the manifestation of wear. However, the easiest method of measuring wear is weighing the test pieces before and after testing.

Provision is made for zeroing the dynamometer and the integrator. The adjustments are made separately with the machine running and the test pieces disengaged. Dynamometer accuracy can be checked by applying a known moment to the lower shaft and noting the dynamometer indication. The machine is stopped for this check, and a lever arm and weight are used to provide the moment.

The machine is fitted with a diagram-recorder which provides a curve of torsional moment vs. revolutions. The operator is offered the choice of 160 or 8000 revolutions per centimeter of paper advance. Because of this limited choice this accessory was not used in the author's tests.

APPENDIX II

HEAT TREATMENT OF THE SPECIMENS

The discs were first placed in an electric furnace at 810°C. (1490°F.) for one hour after which they were removed and allowed to cool in still air for about 25 minutes. No effort was made to control the atmosphere of the furnaces, and thick scale formed on the red hot metal as they cooled. At the end of the cooling period the specimens were inserted into a furnace at 780°C. (1435°F.) where they remained for 30 minutes. They were then quenched for about ten seconds in water at room temperature. After further cooling for about ten minutes in air, the discs entered the tempering furnace. Slack quenching as outlined above was used to prevent cracking of the high-carbon test pieces and was patterned after the method employed by Rosenberg [11] at the National Bureau of Standards. An indication of the amount of heat remaining in the specimens when removed from the quenching bath may be given by the fact that the surface water quickly boiled away. No trouble with cracking of specimens was encountered.

The above procedure was followed for all of the specimens. A batch method was employed to save time. Specimen Nos. 1 through 40 were made up into four batches of ten discs each and were labeled A, B, C, and D. Each of these sets was made by running steel wire through the center holes. Heavy steel washers with outer diameters of $1\frac{1}{4}$ to $1\frac{1}{2}$ in. separated the specimens.

In general, this preparation of sets of specimens left a good deal to be desired. The flexible wire permitted sagging, and uneven heating and cooling probably occurred as a result. When Batch A was removed

from the second furnace for quenching, the wire parted. The red-hot specimens fell to the laboratory floor and started ten small fires which were extinguished with water from the quenching bath. This batch was reassembled, and its heat treatment was begun anew.

Specimens Nos. 41 through 87 were assembled into three sets of 12 discs each and into one set of 11 discs. Named E, F, G, and H, these batches were assembled on fixtures manufactured from $\frac{1}{4}$ in. steel rod. Each fixture contained a straight section about seven inches long. On one end a circular hook was formed to retain the specimens, and on the other end threads were cut. The rod was inserted through the center holes of the specimens using washers for separators as before. A nut was added to the assembly and turned up tight to make a rigid and easily handled mass of metal. It should be noted that because of the washers, the outer portions of the flat sides of the discs were free from contact with other metal.

The tempering of each batch of material followed the slack quenching by about ten minutes. Time of tempering was one hour for all eight batches, and a tempering temperature of 260°C. (500°F.) was used for the first seven batches. Batch H was disassembled when cool, and ten of its specimens were tempered individually using a variety of temperatures.

Throughout all phases of the heat treatment furnace temperatures were checked with a portable thermocouple, for it was determined that some of the installed furnace thermometers were unreliable.

The results of the heat treatment as indicated by hardness measurements are shown in the following four pages. It will be noted that specimen Nos. 41 through 64 of Batches E and F had a fairly uniform

hardness. However, many specimens of the other sets had a wide variation of hardness measurements.

Specimen Number	Hardness Measurements Rockwell C	Average Hardness Rockwell C
1	47, 55, 56, 57½, 57½	54½
2	58½, 57½, 58½	58
3	58, 59, 57½, 57½, 58½	58
4	57½, 41, 56½, 57½, 58, 58½, 40, 39	51
5	54, 48, 60	54
6	spoiled in grinding	-
7	52, 58½, 56½, 58½	56
8	58, 38, 44, 58½	47
9	57, 58, 56, 42	53
10	57, 59, 57, 45, 52, 41	52
11	58, 58, 57, 54	57
12	56½, 57½, 37, 36, 43	46
13	52, 41, 52	48
14	38, 39, 55, 49, 40, 39	43
15	38, 41, 41, 51, 39, 37½, 38½	41
16	40, 39, 39	39
17	38, 45, 58, 38, 55	47
18	41, 50, 39, 40	42½
19	39, 43	41
20	49, 42, 57	49½
21	57½, 58½, 58½, 56½, 40, 45, 50	52
22	39, 43, 48½, 50, 49½, 46, 48	46
23	56, 55, 55½, 55½, 53, 53, 54	54½
24	50, 50, 55, 55, 57, 50, 49, 54	52½
25	55½, 56½, 38, 56, 40, 58, 58, 58	52½

Specimen Number	Hardness Measurements Rockwell C	Average Hardness Rockwell C
26	49, 39, 36, 40	41
27	$49\frac{1}{2}$, $37\frac{1}{2}$, 46, 48, 41, $41\frac{1}{2}$, 48, 48, 48	45
28	47, 40, $41\frac{1}{2}$, $41\frac{1}{2}$, 46, 41	43
29	42, $48\frac{1}{2}$, 46, 42, $39\frac{1}{2}$, $47\frac{1}{2}$	44
30	$57\frac{1}{2}$, 57, 58, $56\frac{1}{2}$, $58\frac{1}{2}$, $58\frac{1}{2}$	$57\frac{1}{2}$
31	$57\frac{1}{2}$, 57, 57, 57, 40, 58, $57\frac{1}{2}$	55
32	44, $45\frac{1}{2}$, $51\frac{1}{2}$, 48, 48, 42	$46\frac{1}{2}$
33	46, 45, 45, 39, 39, 43	43
34	47, 45, 39, 41, 40, 45	43
35	47, 39, 37, $39\frac{1}{2}$, $41\frac{1}{2}$, $41\frac{1}{2}$, $41\frac{1}{2}$	41
36	44, 52, $36\frac{1}{2}$, $36\frac{1}{2}$, 48	$43\frac{1}{2}$
37	$37\frac{1}{2}$, 39, 57, 39	43
38	$35\frac{1}{2}$, $45\frac{1}{2}$, 48, $47\frac{1}{2}$, 40, 40, 38	42
39	$56\frac{1}{2}$, 58, $57\frac{1}{2}$, $57\frac{1}{2}$, 58, 57, 58	$57\frac{1}{2}$
40	$55\frac{1}{2}$, $55\frac{1}{2}$, 58, 53, 56, $53\frac{1}{2}$, 56	$55\frac{1}{2}$
41	$57\frac{1}{2}$, 56, $58\frac{1}{2}$, 58, 56, 58	$57\frac{1}{2}$
42	58, 58, 58, 57, 58, 58	58
43	56, $57\frac{1}{2}$, 58, $57\frac{1}{2}$, $56\frac{1}{2}$	57
44	$56\frac{1}{2}$, $56\frac{1}{2}$, 56, 54, 55	$55\frac{1}{2}$
45	57, $57\frac{1}{2}$, 57, 56, $57\frac{1}{2}$	57
46	57, 57, 54, 57, 57	$56\frac{1}{2}$
47	$53\frac{1}{2}$, $56\frac{1}{2}$, $56\frac{1}{2}$, 57, $56\frac{1}{2}$	56
48	$57\frac{1}{2}$, 57, 57, 55, 57	57
49	56, $56\frac{1}{2}$, 56, 56, $56\frac{1}{2}$	56
50	$53\frac{1}{2}$, $56\frac{1}{2}$, 56, 56, $55\frac{1}{2}$	$55\frac{1}{2}$

Specimen Number	Hardness Measurements Rockwell C	Average Hardness Rockwell C
51	57½, 57½, 57, 58, 58	57½
52	53½, 55, 56, 55½, 54½	55
53	55, 57½, 56½, 56½, 58	56½
54	57½, 57, 57, 57½, 58	57½
55	56, 56½, 57, 57, 55	56½
56	53½, 56½, 56, 56½, 56	55½
57	55, 56½, 56½, 55½, 55	55½
58	52½, 55, 55½, 55½, 55	54½
59	57, 56½, 56½, 57, 56½	56½
60	57, 57, 56½, 57, 57	57
61	54½, 55, 56½, 54½, 56	55½
62	57, 56½, 56½, 57, 57	57
63	56, 56, 57, 56½, 56½	56½
64	55½, 56½, 56½, 56, 56	56
65	57, 57, 56½, 57½, 56½, 40	54
66	40, 57, 54, 47, 57	51
67	56, 48, 57, 57, 36, 57	52
68	52, 53, 57, 55, 57	55
69	56, 57, 57, 57, 56, 45, 54, 57	55
70	38½, 53½, 56½, 56½, 57½	52½
71	56½, 59½, 52½, 55, 55½	56
72	42, 57, 56, 56, 57	53½
73	53½, 56½, 57, 57, 56	55
74	56½, 57, 55½, 55, 57	56
75	57, 56½, 56½, 56½, 57, 56½	56½
76	56½, 56½, 57½, 57½, 57	57

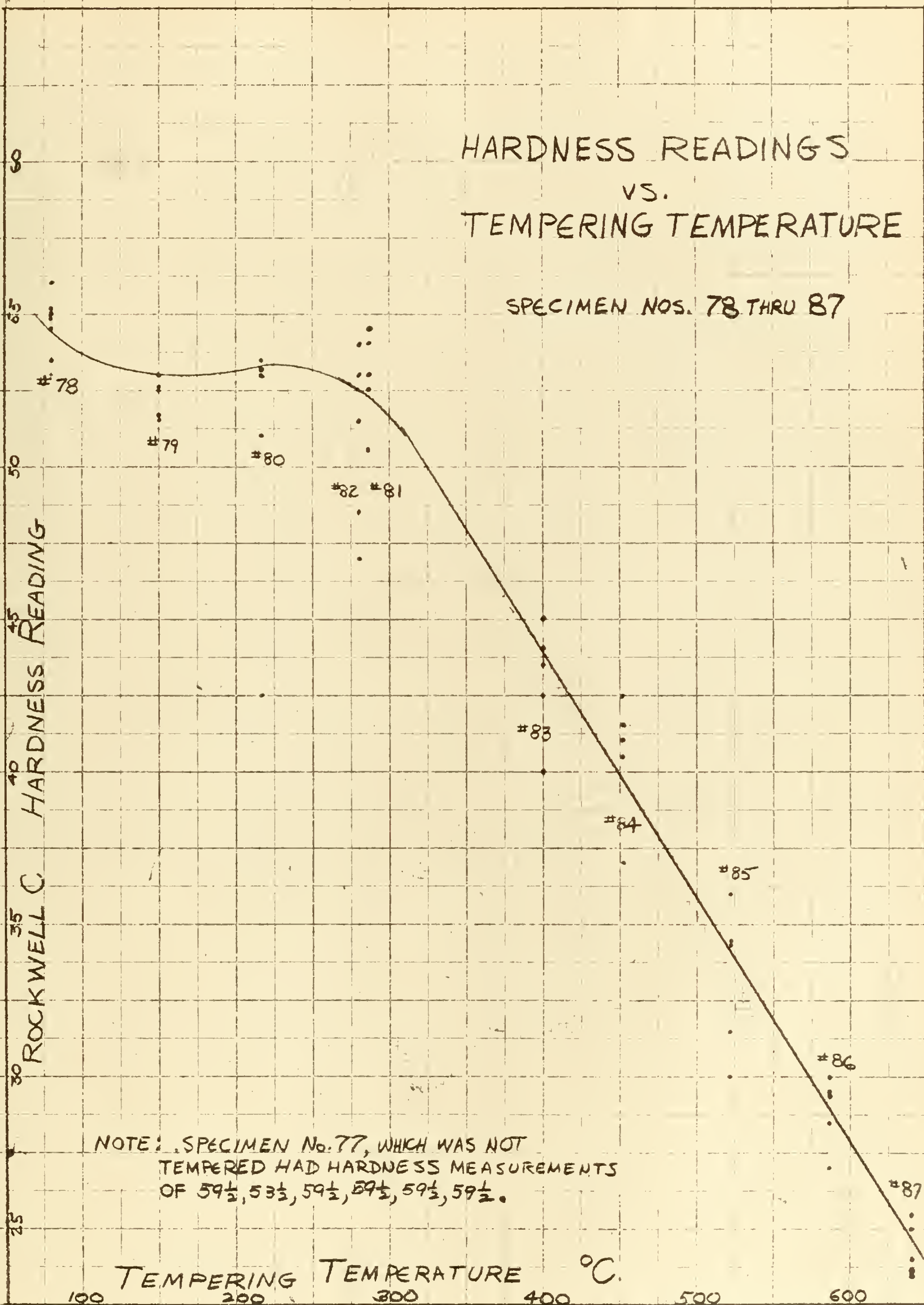


Fig. 15. Hardness readings vs. tempering temperature

APPENDIX III

TEST DATA AND CURVES

Test Run No.	Effective Load Kg	Upper Specimen Number	Lower Specimen Number	Upper Spec Hardness Rockwell C	Lower Spec Hardness Rockwell C
1	60	41	42	$57\frac{1}{2}$	58
2	50	43	44	57	$55\frac{1}{2}$
3	40	45	46	57	$56\frac{1}{2}$
4	30	47	48	56	57
5	14.5	49	50	56	$55\frac{1}{2}$
6	4.5	51	52	$57\frac{1}{2}$	55
7	62.4	53	54	$56\frac{1}{2}$	$57\frac{1}{2}$
8	52.4	55	56	$56\frac{1}{2}$	$55\frac{1}{2}$
9	42.4	57	58	$55\frac{1}{2}$	$54\frac{1}{2}$
10	32.4	59	60	$56\frac{1}{2}$	57
11	22.4	61	62	$55\frac{1}{2}$	57
12	20	63	64	$56\frac{1}{2}$	56
13	60	30	39	$57\frac{1}{2}$	$57\frac{1}{2}$
14	60	87	68	$24\frac{1}{2}$	55
15	60	86	71	29	56
16	60	85	73	$33\frac{1}{2}$	55
17	60	84	74	$40\frac{1}{2}$	56
18	60	83	40	43	$55\frac{1}{2}$
19	60	82	23	50	$54\frac{1}{2}$
20	60	80	11	$50\frac{1}{2}$	57
21	60	79	7	52	56
22	20	76	75	57	$56\frac{1}{2}$
23	60	30 *	41 *	$57\frac{1}{2}$	$55\frac{1}{2}$
24	60	39 *	42 *	$57\frac{1}{2}$	56

* Previously tested specimen reground to 1.985 in. and reused.

Run 1
Load 60 kg
Upper #41 Rc $57\frac{1}{2}$
Lower #42 Rc 58

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	144.1588	145.2320	-	-
250	525	-	-	-	-
600	1,537	-	-	-	-
750	2,265	-	-	-	-
1,000	4,147	-	-	-	-
2,000	10,987	-	-	-	-
3,000	17,910	-	-	-	-
4,000	24,885	-	-	-	-
5,000	32,020	144.1375	145.2263	.0213	.0057
10,000	68,265	144.1195	145.2067	.0393	.0253
15,000	104,389	144.1027	145.1864	.0561	.0456
20,000	140,827	144.0856	145.1702	.0732	.0618
25,000	177,612	144.0699	145.1550	.0889	.0770
30,000	215,107	144.0530	145.1438	.1058	.0882
35,000	252,097	144.0366	145.1322	.1222	.0998
40,000	290,067	144.0199	145.1213	.1389	.1107
45,000	327,823	144.0042	145.1106	.1546	.1114
50,000	366,230	143.9856	145.0912	.1732	.1408

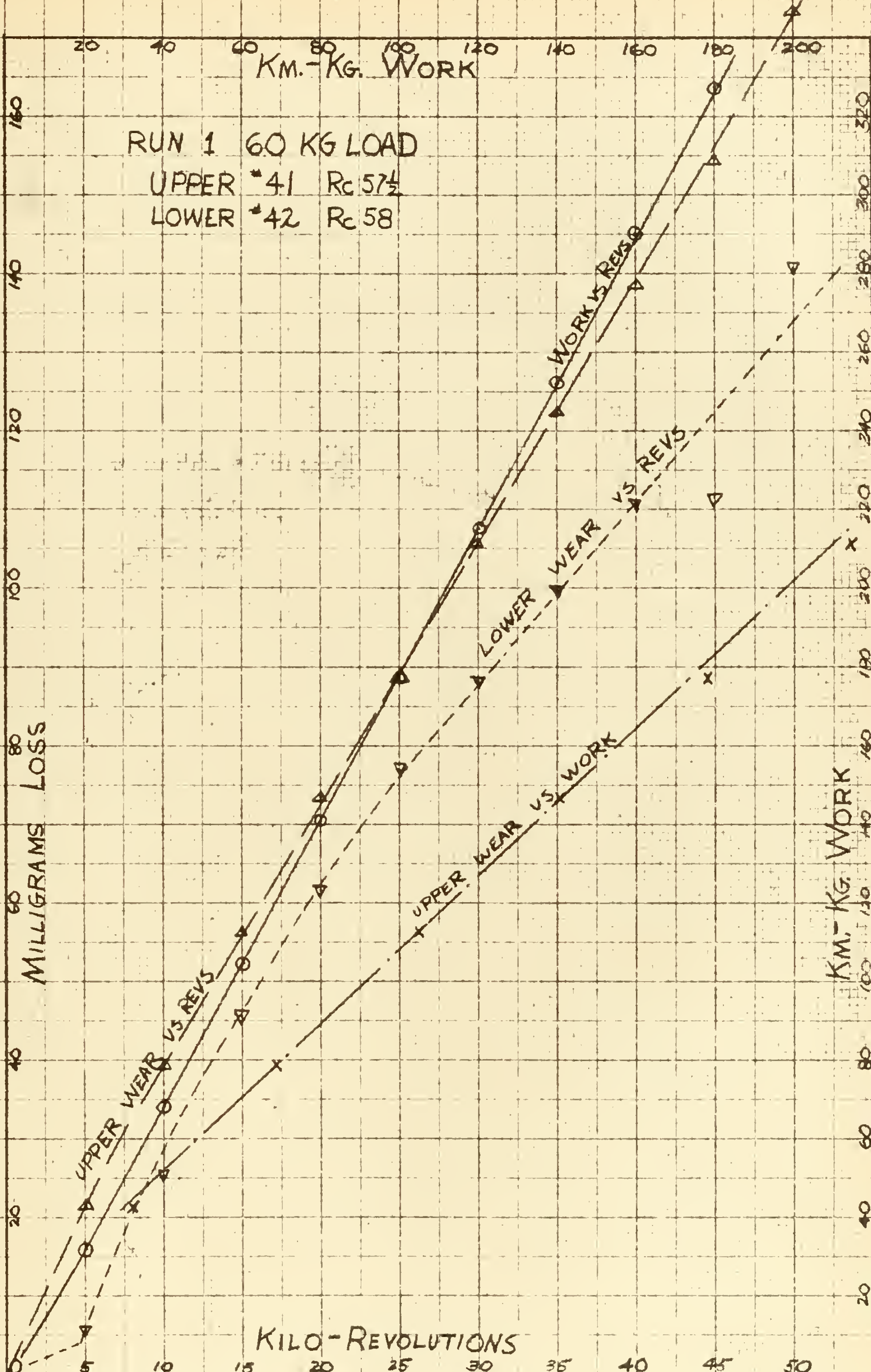


Fig. 16. Curves of results for Run 1 at 60 kg. load

Run 2
Load 50 kg
Upper #43 Rc 57
Lower #44 Rc $55\frac{1}{2}$

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	145.2349	144.9629	-	-
200	372	-	-	-	-
400	672	-	-	-	-
500	858	-	-	-	-
600	1,131	-	-	-	-
700	1,455	-	-	-	-
800	1,927	-	-	-	-
900	2,483	-	-	-	-
1,000	3,053	-	-	-	-
1,500	5,963	-	-	-	-
2,500	11,685	-	-	-	-
5,000	25,944	145.2147	144.9551	.0202	.0078
10,000	55,759	145.1983	144.9384	.0366	.0235
15,000	86,019	145.1824	144.9195	.0525	.0434
20,000	116,751	145.1685	144.9030	.0664	.0599
25,000	147,295	145.1554	144.8869	.0795	.0760
30,000	178,158	145.1425	144.8772	.0924	.0857
40,000	242,832	145.1127	144.8475	.1222	.1154
50,000	308,245	145.0838	144.8225	.1511	.1404

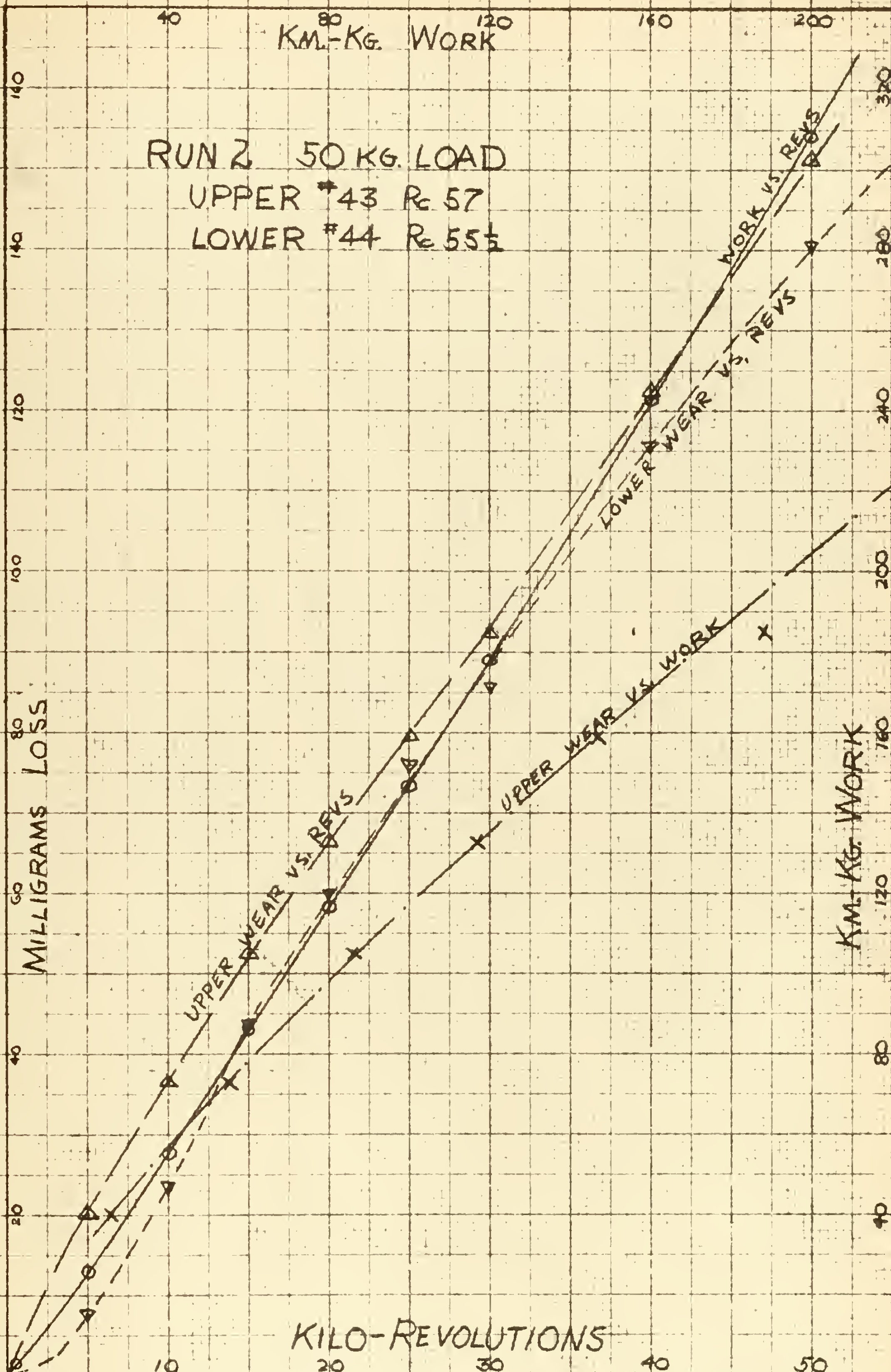


Fig. 17. Curves of results for Run 2 at 50 kg. load

Run 3
Load 40 kg
Upper #45 Rc 57
Lower #46 Rc $56\frac{1}{2}$

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	144.2113	145.2658	-	-
200	266	-	-	-	-
300	354	-	-	-	-
400	454	-	-	-	-
500	565	-	-	-	-
600	690	-	-	-	-
700	835	-	-	-	-
800	1,012	-	-	-	-
900	1,428	-	-	-	-
1,000	1,530	-	-	-	-
1,500	3,612	-	-	-	-
2,500	8,135	-	-	-	-
5,000	19,553	144.1987	145.2657	.0126	.0001
10,000	43,930	144.1852	145.2509	.0261	.0149
15,000	68,958	144.1735	145.2381	.0378	.0277
20,000	94,101	144.1611	145.2268	.0502	.0390
25,000	118,980	144.1489	145.2133	.0624	.0515
30,000	143,919	144.1385	145.2046	.0728	.0612
50,000	252,852	144.0913	145.1649	.1200	.1009

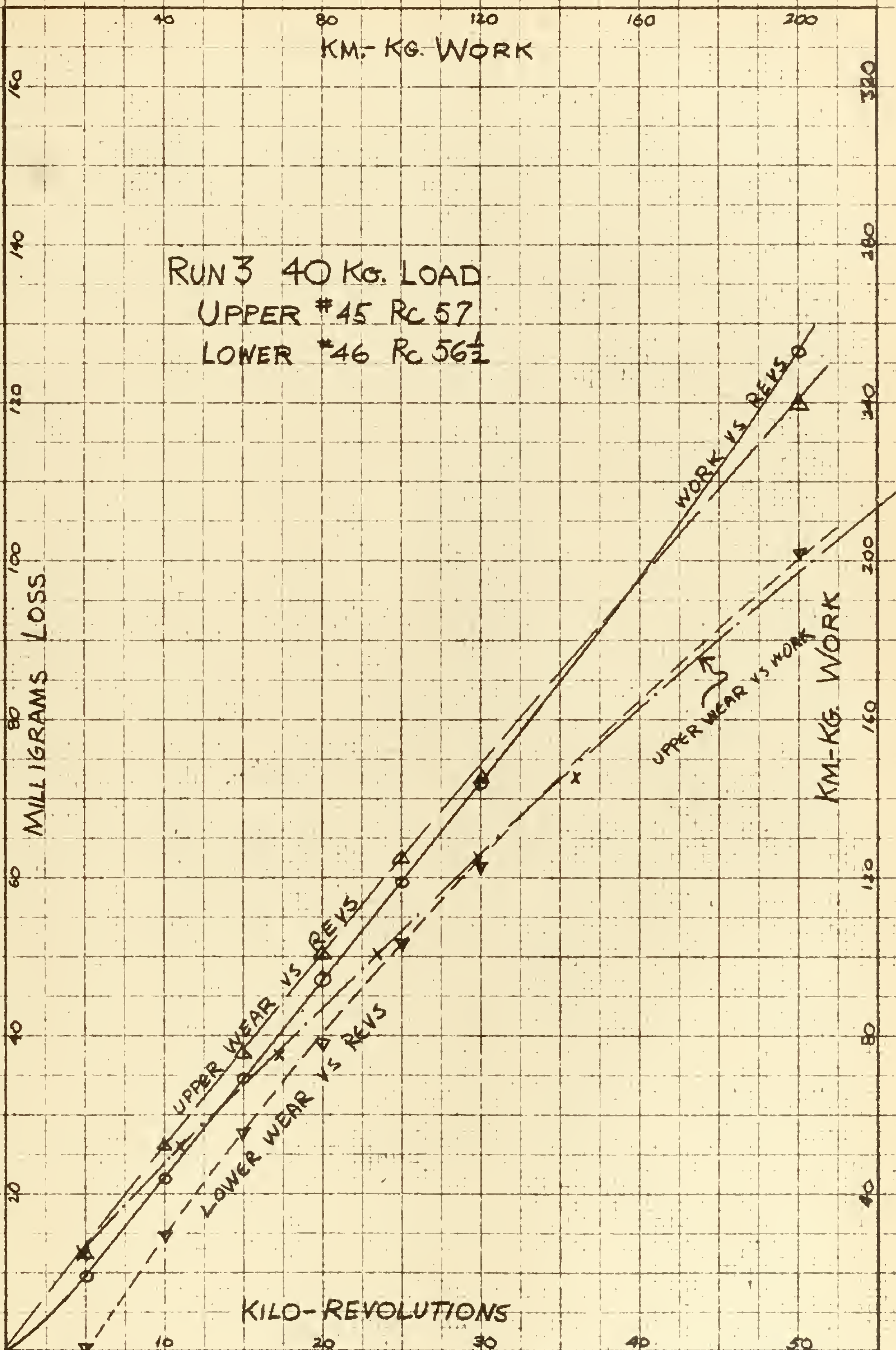


Fig. 18. Curves of results for Run 3 at 40 kg. load

Run 4
Load 30 kg
Upper #47 Rc 56
Lower #48 Rc 57

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	144.9986	145.2352	-	-
100	170	-	-	-	-
200	238	-	-	-	-
300	313	-	-	-	-
400	395	-	-	-	-
500	487	-	-	-	-
700	720	-	-	-	-
800	859	-	-	-	-
900	1,024	-	-	-	-
1,000	1,210	-	-	-	-
1,500	2,660	-	-	-	-
2,000	4,409	-	-	-	-
2,500	6,175	-	-	-	-
3,000	7,980	-	-	-	-
4,000	11,610	-	-	-	-
5,000	15,251	144.9878	145.2348	.0108	.0004
10,000	34,028	144.9762	145.2260	.0224	.0092
15,000	53,280	144.9659	145.2186	.0327	.0166
20,000	72,788	144.9559	145.2116	.0427	.0236
25,000	92,052	144.9462	145.2033	.0524	.0319
30,000	111,308	144.9392	145.1988	.0594	.0364
50,000	192,998	144.8996	145.1641	.0990	.0711

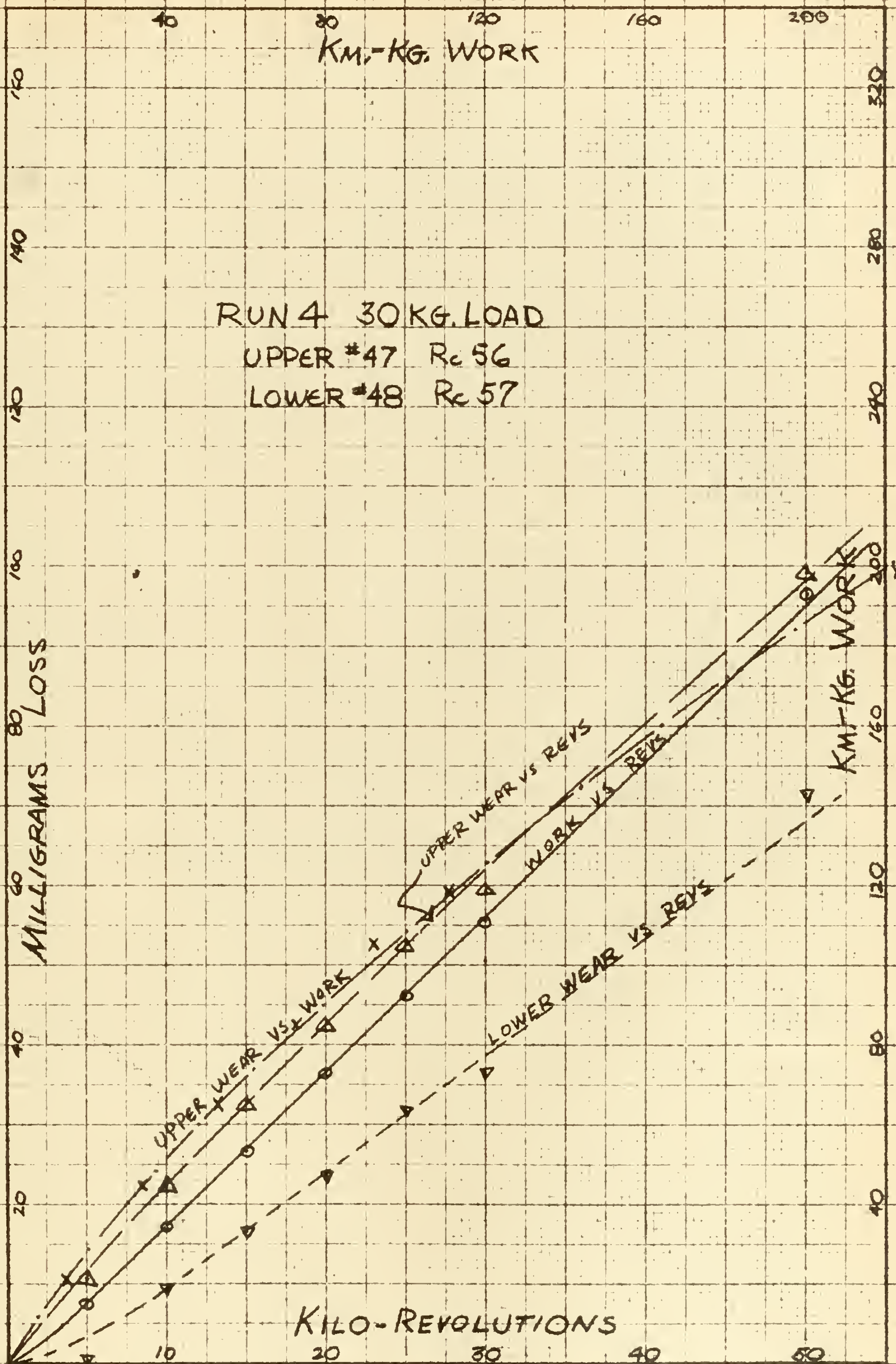


Fig. 19. Curves of results for Run 4 at 30 kg. load

Run 5
Load 14.5 kg
Upper #49 Rc 56
Lower #50 Rc 55½

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	144.1522	144.7685	-	-
200	136	-	-	-	-
400	221	-	-	-	-
500	266	-	-	-	-
600	308	-	-	-	-
800	400	-	-	-	-
1,000	499	-	-	-	-
1,500	780	-	-	-	-
2,500	1,630	-	-	-	-
5,000	6,540	144.1523	144.7695	Gain	Gain
10,000	17,162	144.1513	144.7682	.0009	.0003
15,000	28,544	144.1512	144.7677	.0010	.0008
20,000	41,154	144.1411	144.7648	.0111	.0037
25,000	54,108	144.1355	144.7606	.0167	.0079
30,000	67,891	144.1301	144.7569	.0221	.0116
50,000	121,825	144.1055	144.7440	.0467	.0245
65,000	162,265	144.0848	144.7301	.0674	.0384
115,000	589,118	144.0301	144.7183	.1221	.0502
120,000	602,083	-	-	-	-
125,000	615,673	144.0165	144.7110	.1357	.0575
135,000	643,146	144.0041	144.7077	.1481	.0608

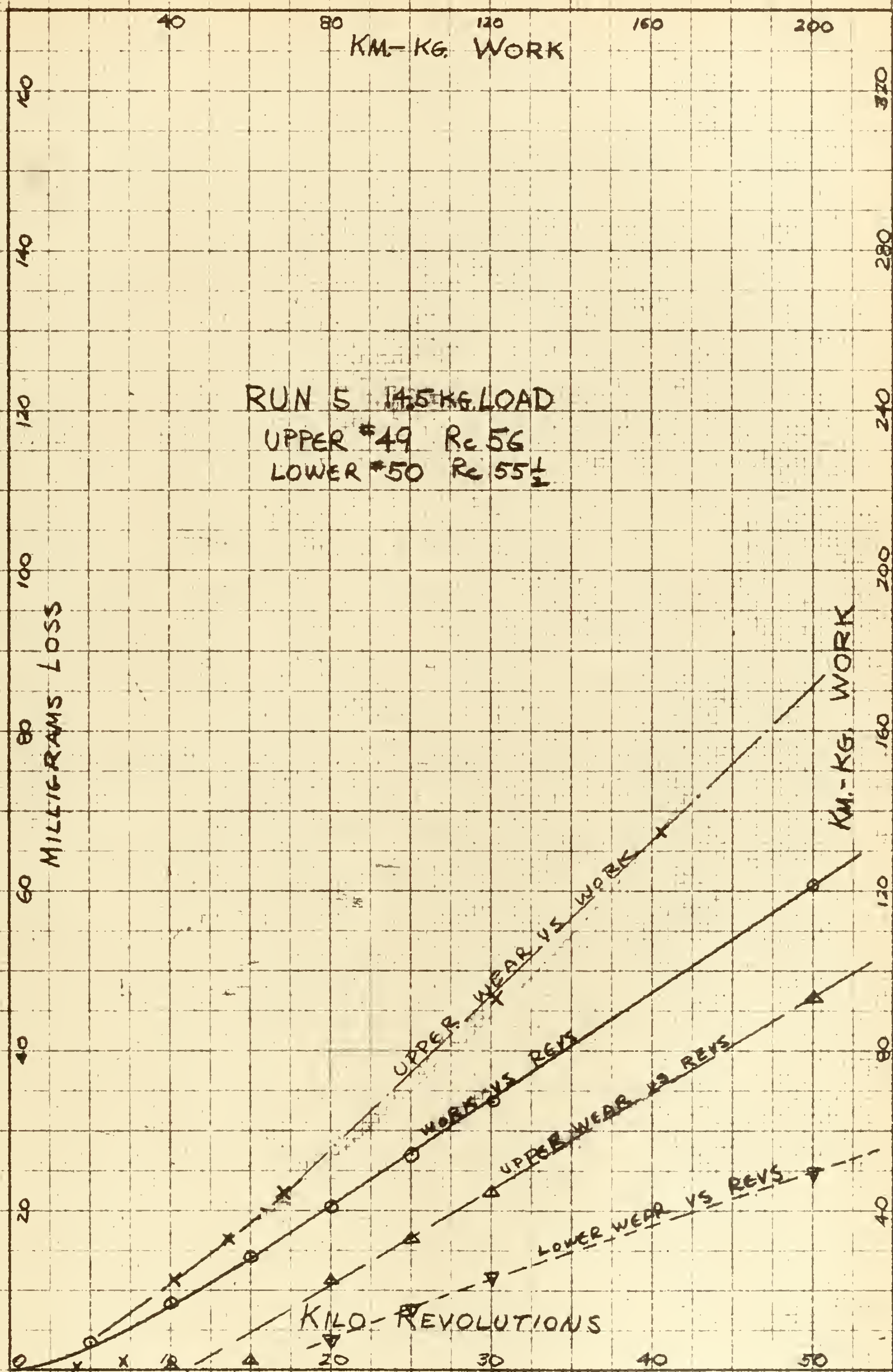


Fig. 20. Curves of results for Run 5 at 14.5 kg. load

Run 6
Load 4.5 kg
Upper #51 Rc $57\frac{1}{2}$
Lower #52 Rc 55

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	145.1968	144.7081	-	-
100	78	-	-	-	-
200	105	-	-	-	-
300	132	-	-	-	-
400	158	-	-	-	-
500	185	-	-	-	-
800	273	-	-	-	-
1,000	340	-	-	-	-
2,000	860	-	-	-	-
3,000	1,965	-	-	-	-
5,000	4,472	145.1973	144.7079	Gain	.0002
10,000	9,468	145.1969	144.7078	Gain	.0003
15,000	15,078	145.1965	144.7074	.0003	.0007
20,000	20,585	145.1968	144.7071	.0000	.0010
25,000	26,211	145.1965	144.7061	.0003	.0020
30,000	32,482	145.1962	144.7061	.0006	.0020
40,000	45,382	145.1947	144.7047	.0021	.0034
125,000	158,247	145.1423	144.6979	.0545	.0102
145,000	184,824	145.1289	144.6969	.0679	.0112
185,000	238,028	145.1027	144.6939	.0941	.0142
200,000	257,976	145.0937	144.6953	.1031	.0128

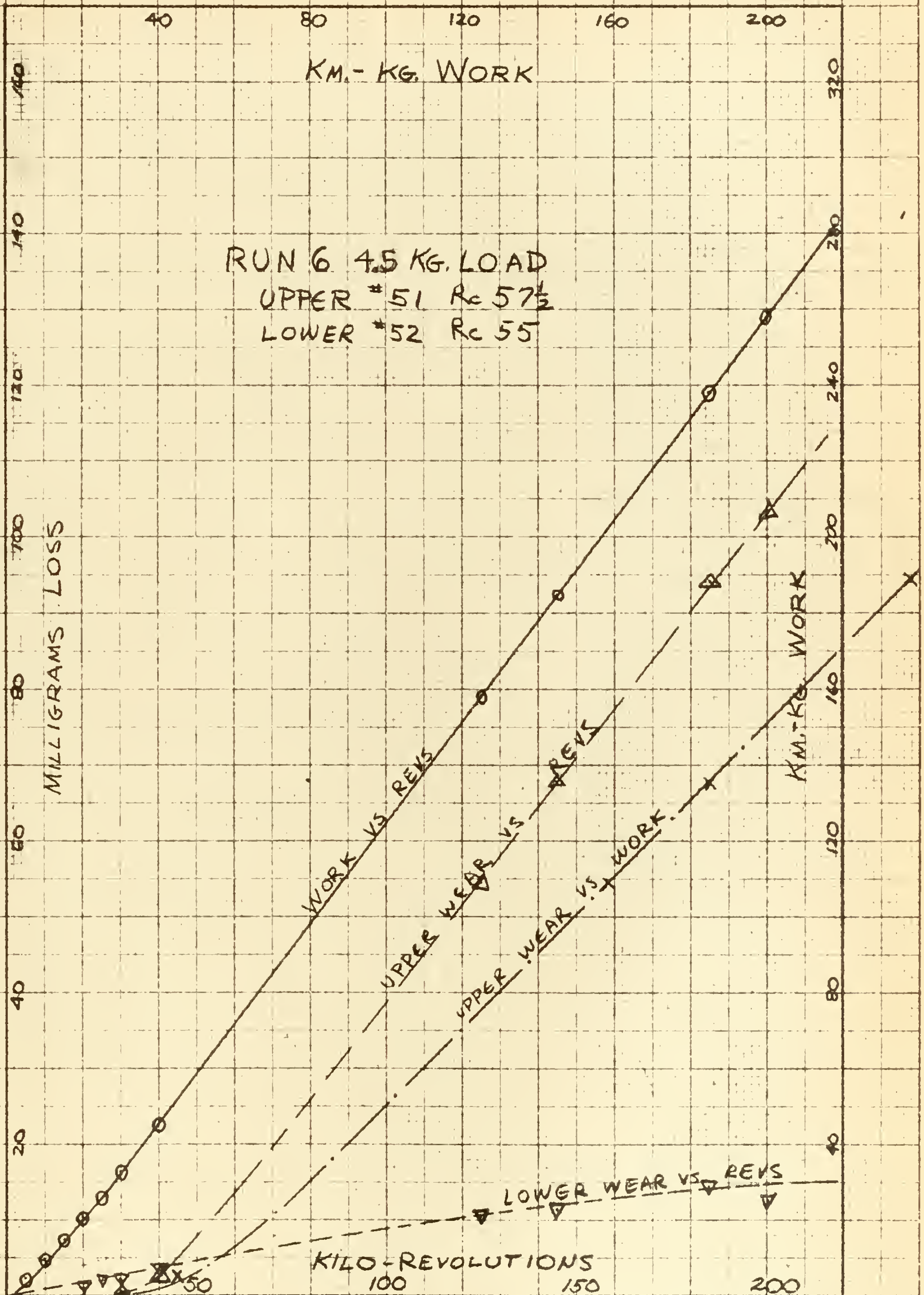


Fig. 21. Curves of results for Run 6 at 4.5 kg. load

Run 7
Load 62.4 kg
Upper #53 Rc $56\frac{1}{2}$
Lower #54 Rc $57\frac{1}{2}$

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	144.8408	145.2365	-	-
5,000	32,537	144.8153	145.2275	.0255	.0090
10,000	70,092	144.7982	145.2082	.0426	.0283
15,000	108,045	144.7813	145.1905	.0595	.0460
20,000	146,119	144.7649	145.1752	.0759	.0613
25,000	184,313	144.7496	145.1577	.0912	.0788
30,000	223,887	144.7324	145.1433	.1084	.0932
40,000	301,281	144.7004	145.1214	.1404	.1151
50,000	379,302	144.6682	145.1003	.1726	.1362

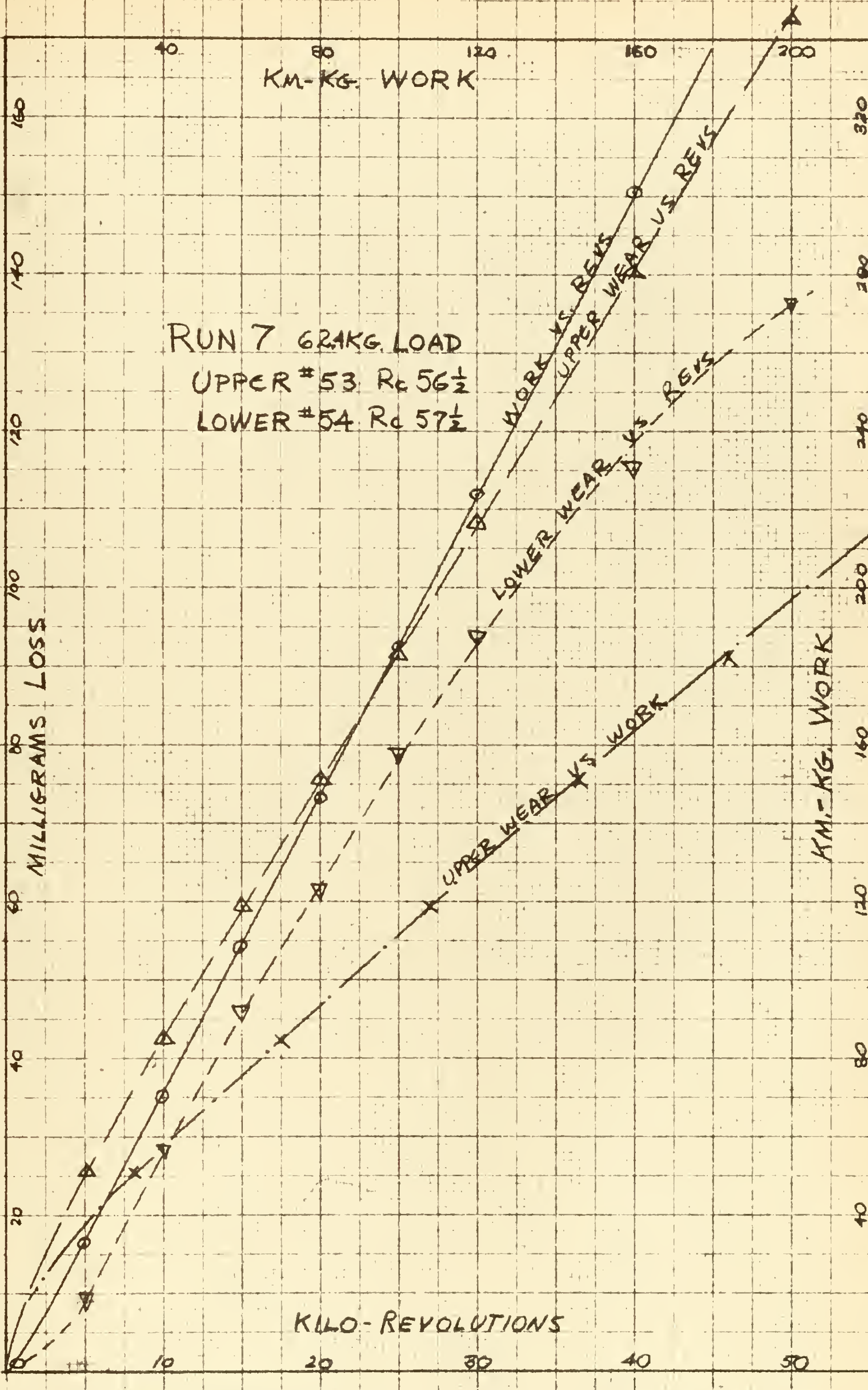


Fig. 22. Curves of results for Run 7 at 62.4 kg. load

Run 8
Load 52.4 kg
Upper #55 Rc 56½
Lower #56 Rc 55½

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	144.9248	145.2161	-	-
5,000	28,116	144.9066	145.2017	.0182	.0144
10,000	60,167	144.8903	145.1831	.0345	.0330
15,000	92,645	144.8747	145.1628	.0501	.0533
20,000	125,184	144.8605	145.1462	.0643	.0699
25,000	158,484	144.8446	145.1298	.0802	.0863
30,000	192,279	144.8293	145.1218	.0955	.0943
40,000	258,840	144.8009	145.1062	.1239	.1099
50,000	326,441	144.7754	145.0955	.1494	.1206

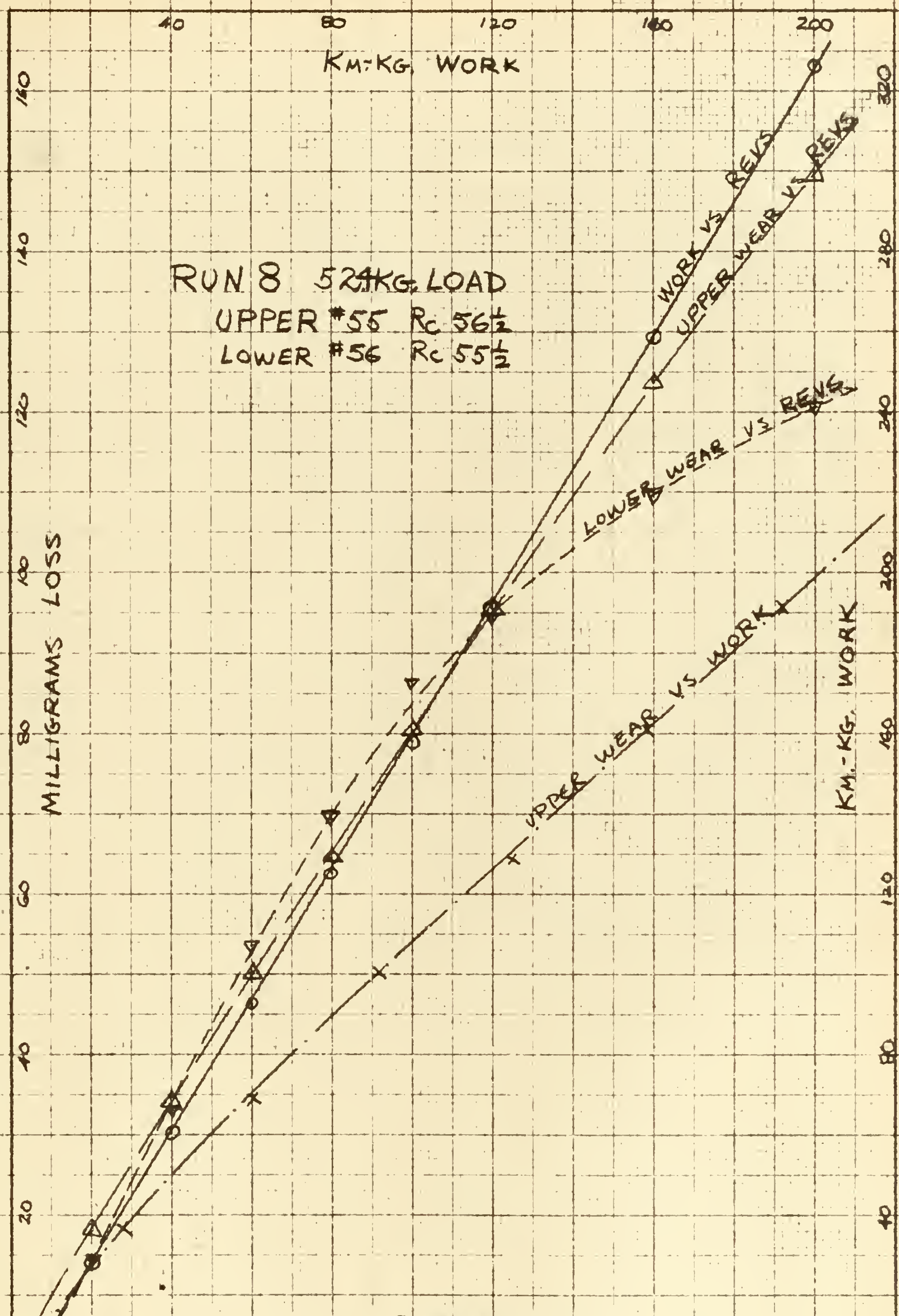


Fig. 23. Curves of results for Run 8 at 52.4 kg load

Run 9
Load 42.4 kg
Upper #57 Rc $55\frac{1}{2}$
Lower #58 Rc $54\frac{1}{2}$

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	145.3526	144.8339	-	-
5,000	23,654	145.3353	144.8342	.0173	Gain
10,000	50,288	145.3193	144.8170	.0333	.0169
15,000	72,915	145.3053	144.7997	.0473	.0342
20,000	104,983	145.2930	144.7884	.0596	.0455
25,000	132,354	145.2799	144.7722	.0727	.0617
30,000	159,699	145.2671	144.7600	.0855	.0739
50,000	270,624	145.2187	144.7313	.1339	.1026

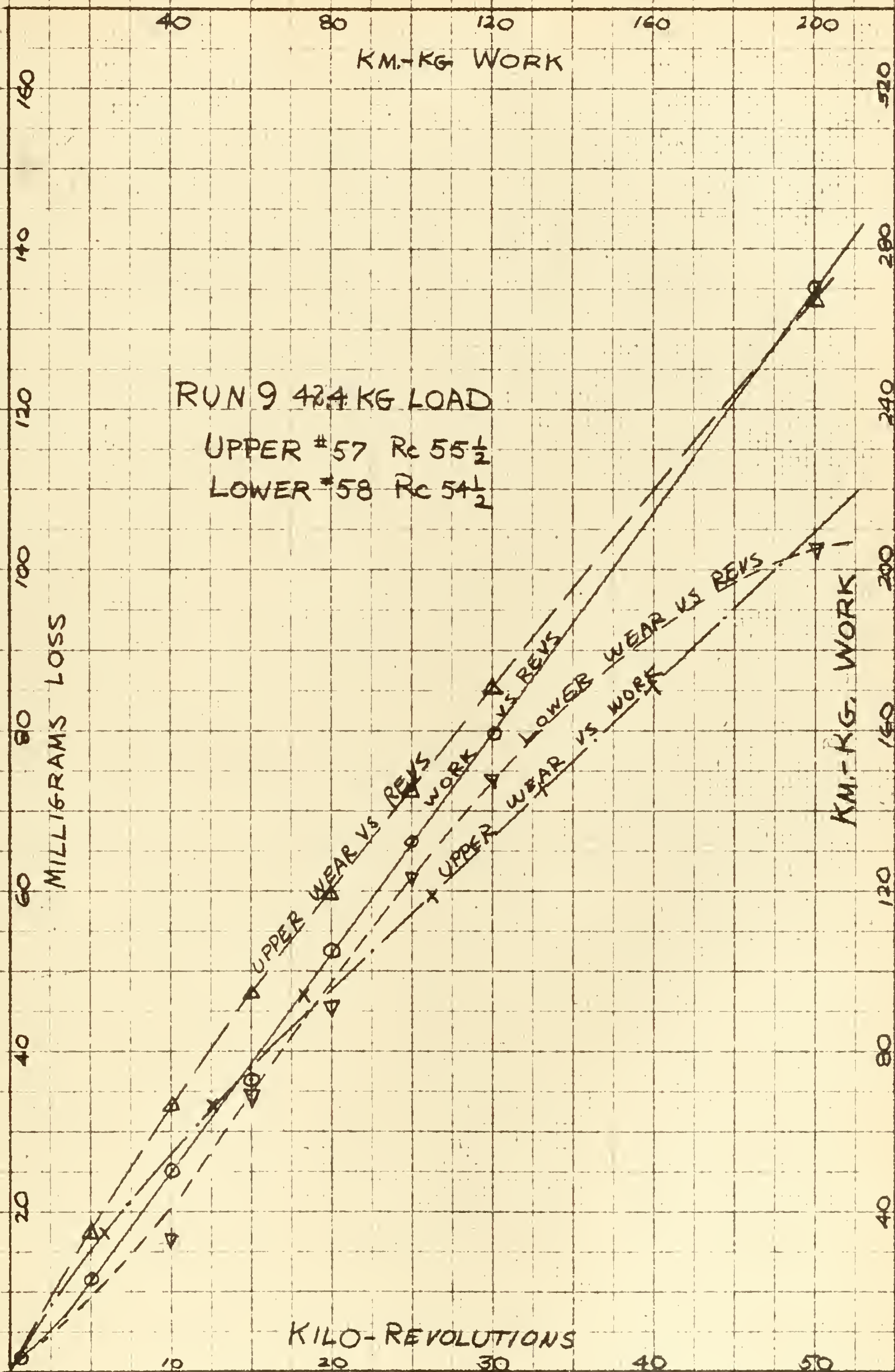


Fig. 24. Curves of results for Run 9 at 42.4 kg. load

Run 10
Load 32.4 kg
Upper #59 Rc $56\frac{1}{2}$
Lower #60 Rc 57

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	145.2725	145.2451	-	-
5,000	15,994	145.2616	145.2455	.0109	Gain
10,000	36,501	145.2486	145.2355	.0239	.0096
15,000	57,487	145.2387	145.2272	.0338	.0179
20,000	78,740	145.2287	145.2172	.0438	.0279
25,000	100,218	145.2191	145.2103	.0534	.0348
30,000	121,725	145.2093	145.2058	.0632	.0393
50,000	208,156	145.1705	145.1884	.1020	.0567

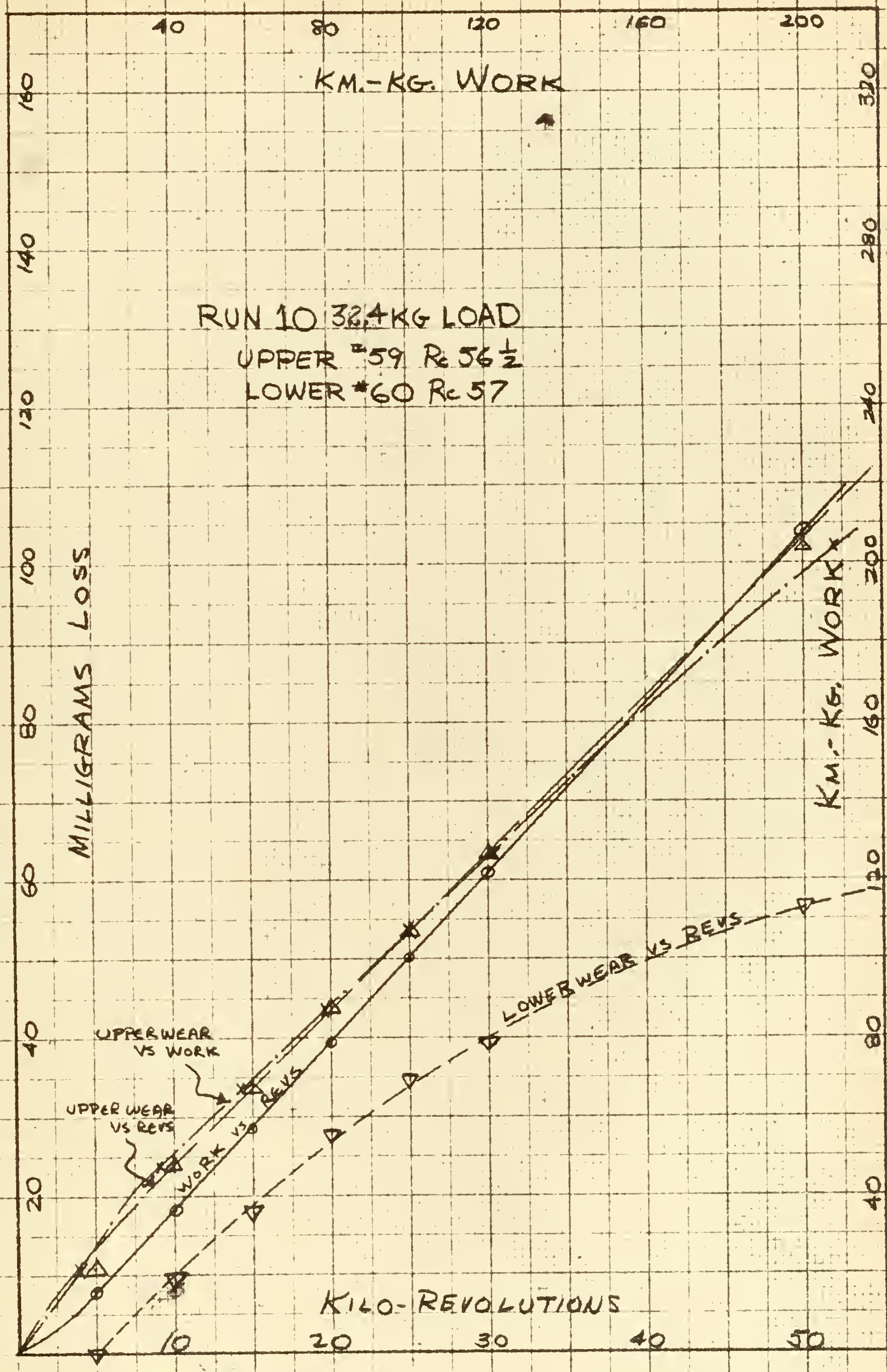


Fig. 25. Curves of results for Run 10 at 32.4 kg. load

Run 11
Load 22.4 kg
Upper #61 Rc $55\frac{1}{2}$
Lower #62 Rc 57

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	144.4912	145.3933	-	-
5,000	12,978	144.4865	145.3944	.0047	Gain
9,000	26,477	144.4783	145.3913	.0129	.0020
15,000	43,844	144.4669	145.3878	.0243	.0055
20,000	64,001	144.4582	145.3873	.0330	.0060
25,000	81,162	144.4502	145.3875	.0410	.0058
30,000	98,241	144.4422	145.3878	.0490	.0055
40,000	131,384	144.4260	145.3856	.0652	.0077
50,000	164,141	144.4096	145.3835	.0816	.0098

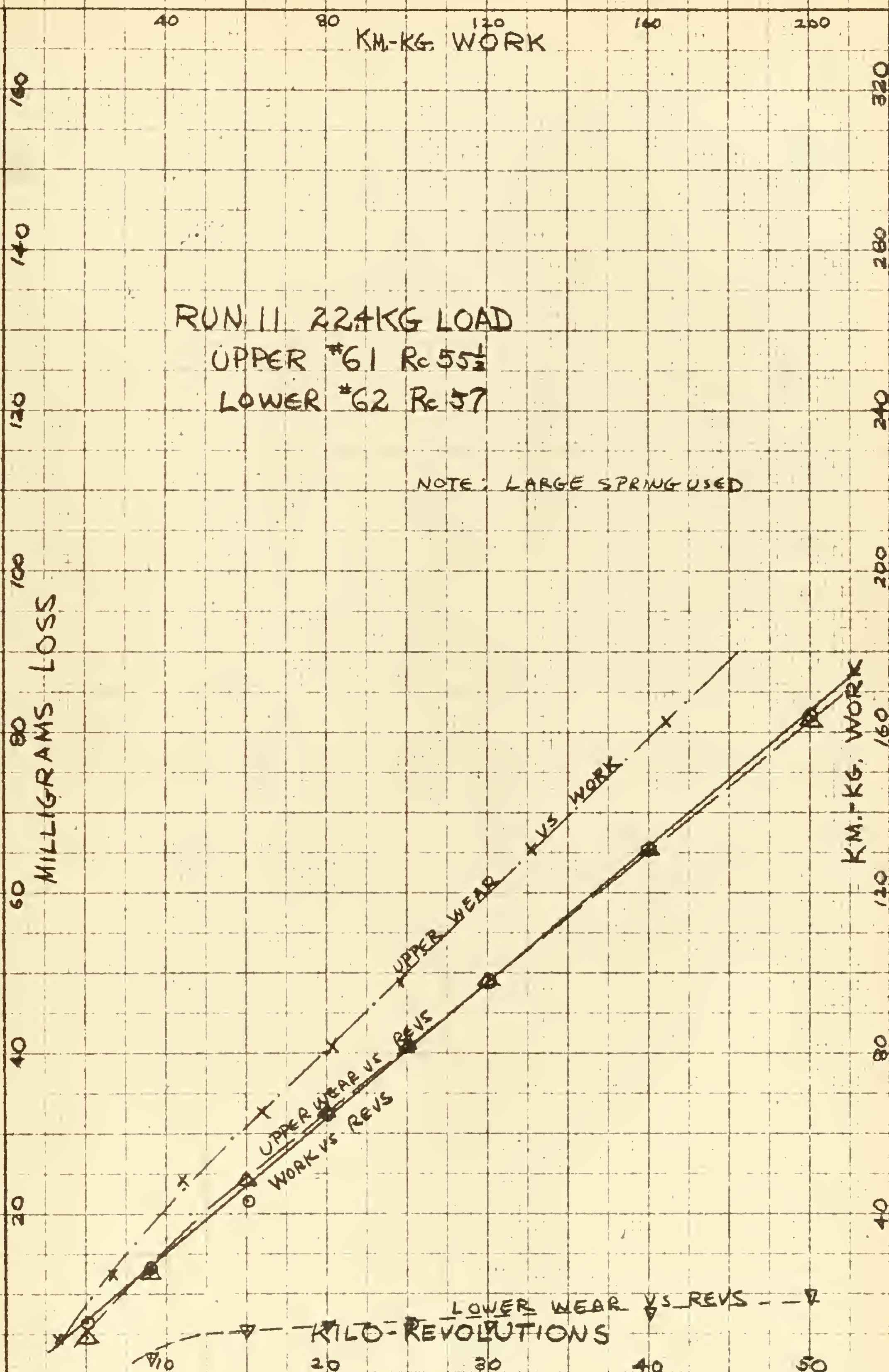


Fig. 26. Curves of results for Run 11 at 22.4 kg. load

Run 12
Load 20 kg
Upper #63 Rc 56½
Lower #64 Rc 56

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	145.1369	144.7569	-	-
5,000	11,607	145.1338	144.7580	.0031	Gain
10,000	26,959	145.1249	144.7542	.0120	.0027
15,000	42,241	145.1166	144.7499	.0203	.0070
20,000	58,019	145.1095	144.7491	.0274	.0078
25,000	73,441	145.1013	144.7466	.0356	.0103
30,000	89,552	145.0935	144.7478	.0434	.0091
35,000	104,715	145.0885	144.7389	.0484	.0180
40,000	120,319	145.0819	144.7353	.0550	.0216
45,000	136,446	145.0739	144.7315	.0630	.0254
50,000	152,253	145.0684	144.7272	.0685	.0297

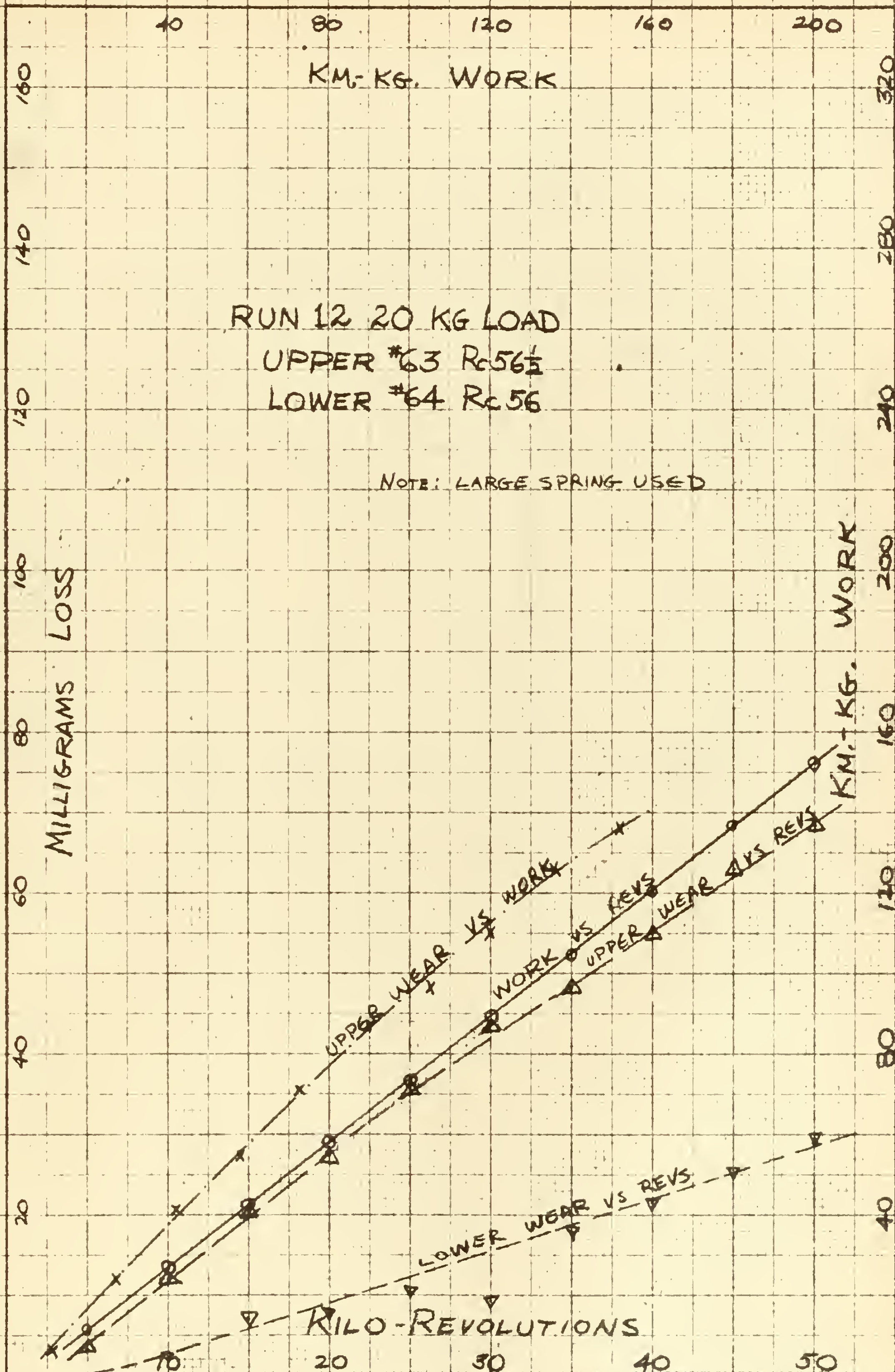


Fig. 27 Curves of results for Run 12 at 20 kg. load

Run 13
Load 60 kg
Upper #30 Rc $57\frac{1}{2}$
Lower #39 Rc $57\frac{1}{2}$

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	144.6391	144.4698	-	-
5,000	31,419	144.6261	144.4677	.0130	.0021
10,000	68,153	144.6113	144.4476	.0278	.0222
15,000	105,687	144.5946	144.4290	.0445	.0408
20,000	144,050	144.5796	144.4173	.0595	.0525
25,000	182,475	144.5647	144.4065	.0744	.0633
30,000	220,567	144.5487	144.3937	.0904	.0761
40,000	296,257	144.5175	144.3739	.1216	.0959
50,000	370,947	144.4867	144.3502	.1524	.1196

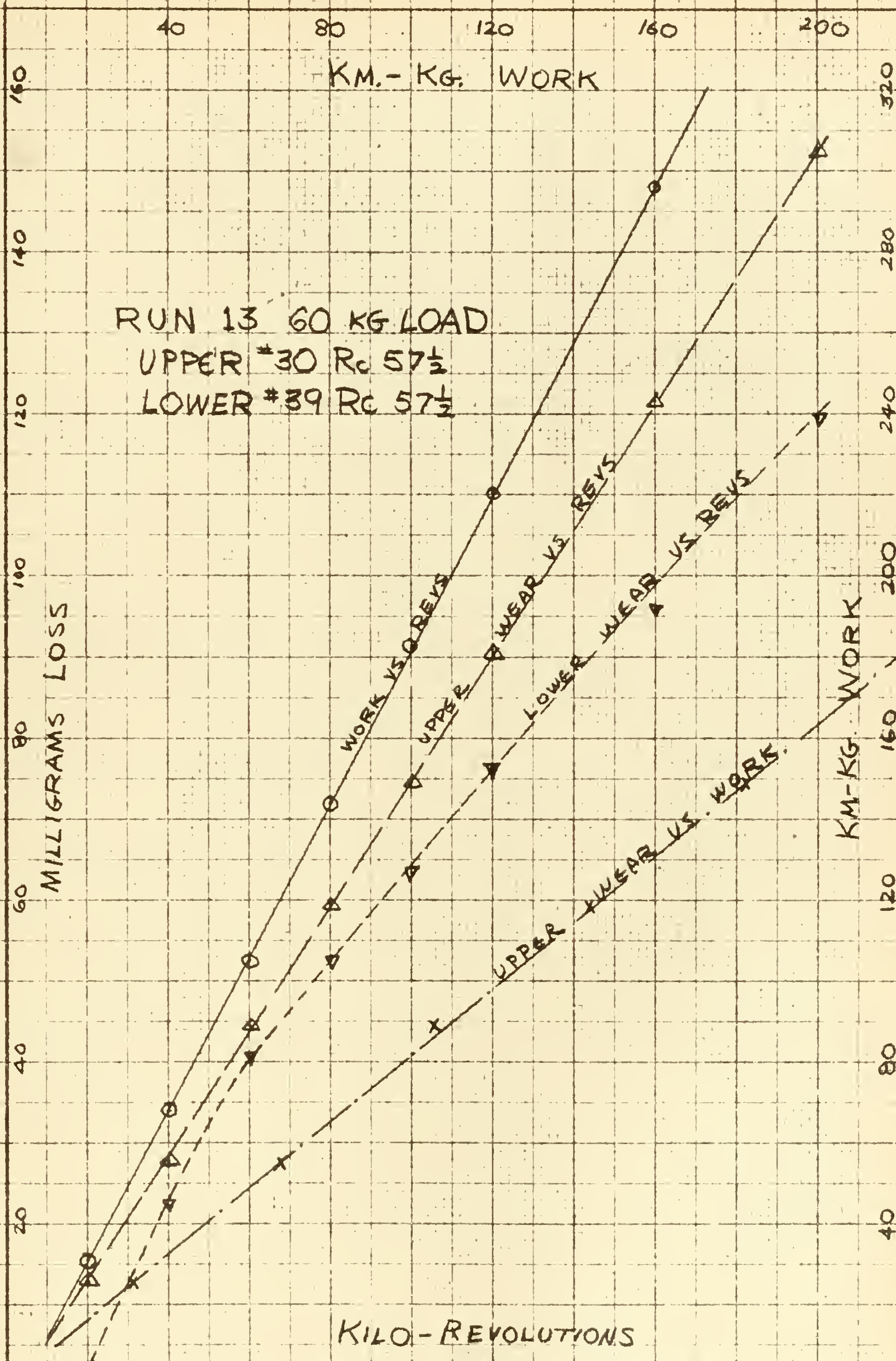


Fig. 28. Curves of results for Run 13 at 60 kg. load

Runs 14 thru 21
Load 60 kg

Run No.	Upper Specimen Number	Upper Hardness Rc	Upper Tempering Temperature	Lower Specimen Number	Lower Hardness Rc	Revs for 100,000 M-Kg
14	87	$24\frac{1}{2}$	640°C	68	55	15,120
15	86	29	586°C	71	56	15,210
16	85	$33\frac{1}{2}$	522°C	73	55	14,940
17	84	$40\frac{1}{2}$	452°C	74	56	15,180
18	83	43	400°C	40	$55\frac{1}{2}$	15,380
19	82	50	280°C	23	$54\frac{1}{2}$	15,430
20	80	$50\frac{1}{2}$	216°C	11	57	17,370 *
21	79	52	150°C	7	56	15,510

* Reading is not necessarily inconsistent. Load was removed briefly when machine threatened to jam.

Run No.	Initial Upper Grams	Final Upper Grams	Upper Grams Lost	Initial Lower Grams	Final Lower Grams	Lower Grams Lost
14	145.0169	144.1913	.8256	144.6806	144.6724	.0082
15	145.0615	144.2147	.8468	144.6812	144.6542	.0270
16	146.2715	146.0455	.2260	144.8361	144.8123	.0238
17	145.1227	144.9351	.1876	144.6406	144.6124	.0282
18	144.8156	144.6403	.1753	144.8523	specimen damaged	
19	144.9608	144.8982	.0626	145.4468	145.4272	.0196
20	144.6213	144.5654	.0559	145.6746	145.6729	.0017
21	144.9821	144.9108	.0713	145.1326	145.1123	.0203

Run 22
Load 20 kg
Upper #76 Rc 57
Lower #75 Rc $56\frac{1}{2}$

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	144.5829	144.6611	-	-
5,000	7,082	144.5817	144.6613	.0012	Gain
10,000	19,780	144.5824	144.6617	.0005	Gain
15,000	34,276	144.5752	144.6595	.0077	.0016
20,000	50,256	144.5655	144.6535	.0174	.0076
25,000	66,500	144.5587	144.6518	.0242	.0093
30,000	83,093	144.5503	144.6500	.0326	.0111
35,000	99,629	144.5425	144.6499	.0404	.0112
40,000	115,608	144.5346	144.6495	.0483	.0116
45,000	131,520	144.5267	144.6479	.0562	.0132
50,000	147,647	144.5194	144.6423	.0635	.0188

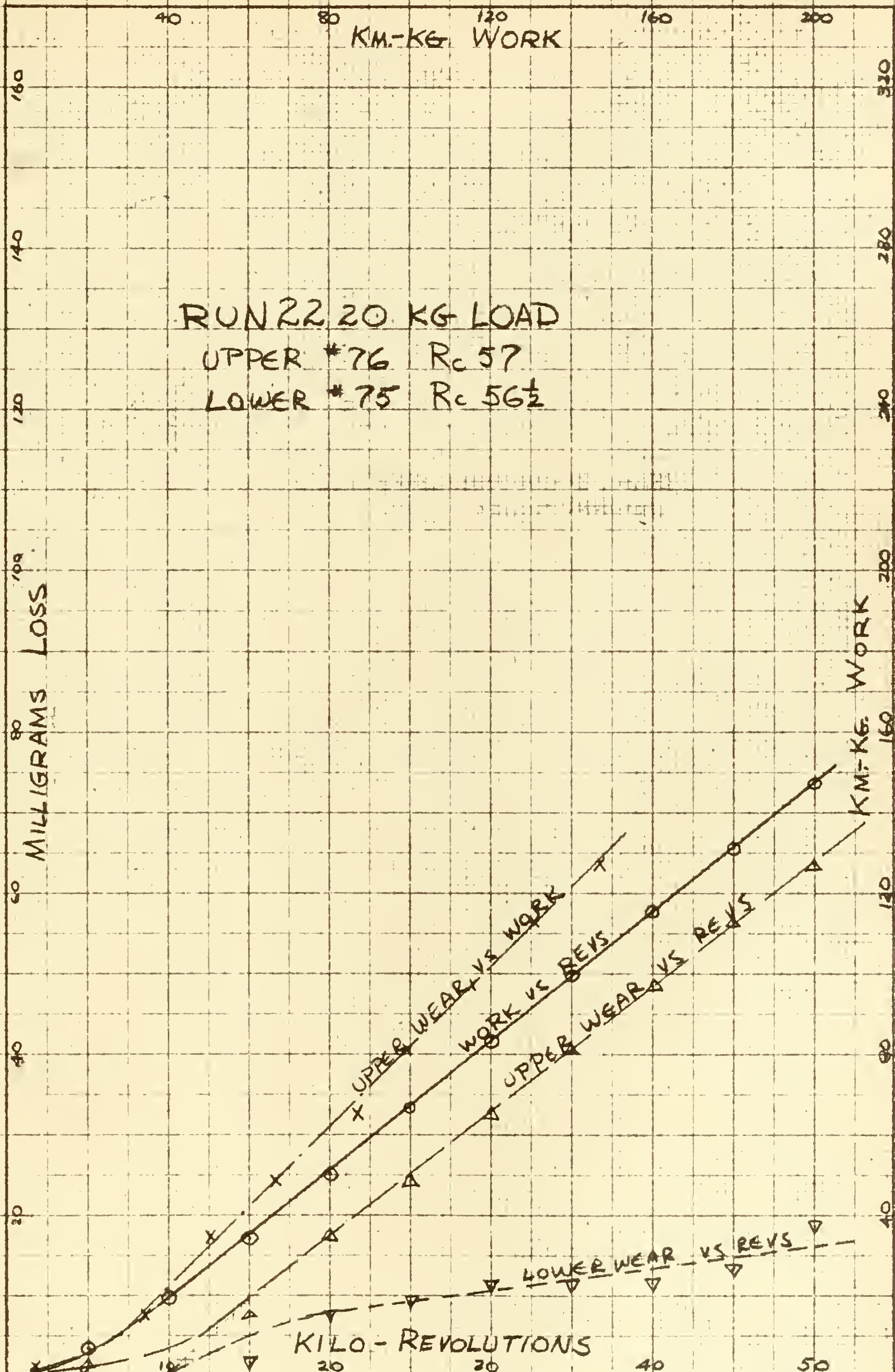


Fig. 29. Curves of results for Run 22 at 20 kg. load

Run 23
Load 60 kg
Upper #30 Rc 57½
Lower #41 Rc 55½

Specimens previously used and reground to 1.985 in.

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	142.4119	141.7948	-	-
5,000	14,834	142.4115	141.7946	.0004	.0002
10,000	48,492	142.3969	141.7836	.0150	.0112
15,000	82,446	142.3841	141.7614	.0278	.0334
20,000	119,957	142.3680	141.7459	.0439	.0489
25,000	158,028	142.3509	141.7329	.0610	.0619
30,000	195,811	142.3350	141.7206	.0769	.0742
35,000	232,962	142.3192	141.7041	.0927	.0907
40,000	271,565	142.3035	141.6943	.1084	.1005
45,000	306,499	142.2862	141.6691	.1257	.1257
50,000	341,163	142.2709	141.6476	.1410	.1472

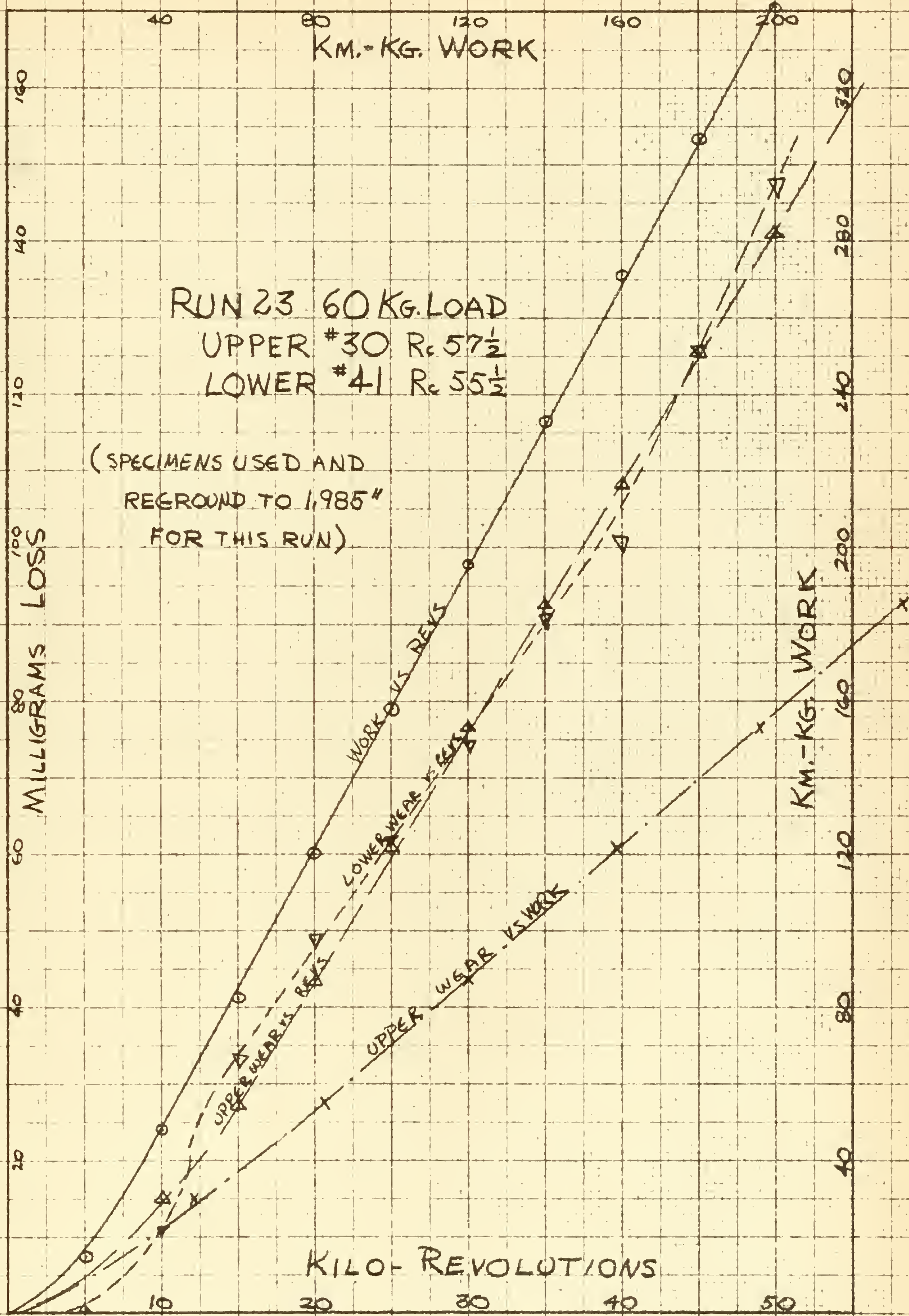


Fig. 30. Curves of results for Run 23 at 60 kg. load

Run 24
Load 60 kg
Upper #39 Rc $57\frac{1}{2}$
Lower #42 Rc 56

Specimens previously used and reground to 1.985 in.

Test Revs	M-Kg Work	Upper Grams	Lower Grams	Upper Loss	Lower Loss
0	0	142.1929	142.9455	-	-
5,000	28,800	142.1813	142.9466	.0116	Gain
10,000	64,380	142.1648	142.9213	.0281	.0242
15,000	100,525	142.1489	142.8989	.0440	.0466
20,000	137,053	142.1337	142.8767	.0592	.0668
25,000	173,998	142.1164	142.8650	.0765	.0805
30,000	212,249	142.1003	142.8516	.0926	.0939
35,000	249,303	142.0841	142.8392	.1088	.1063
40,000	287,055	142.0689	142.8248	.1240	.1207
45,000	324,496	142.0535	142.8127	.1394	.1328
50,000	361,933	142.0387	142.8016	.1542	.1439

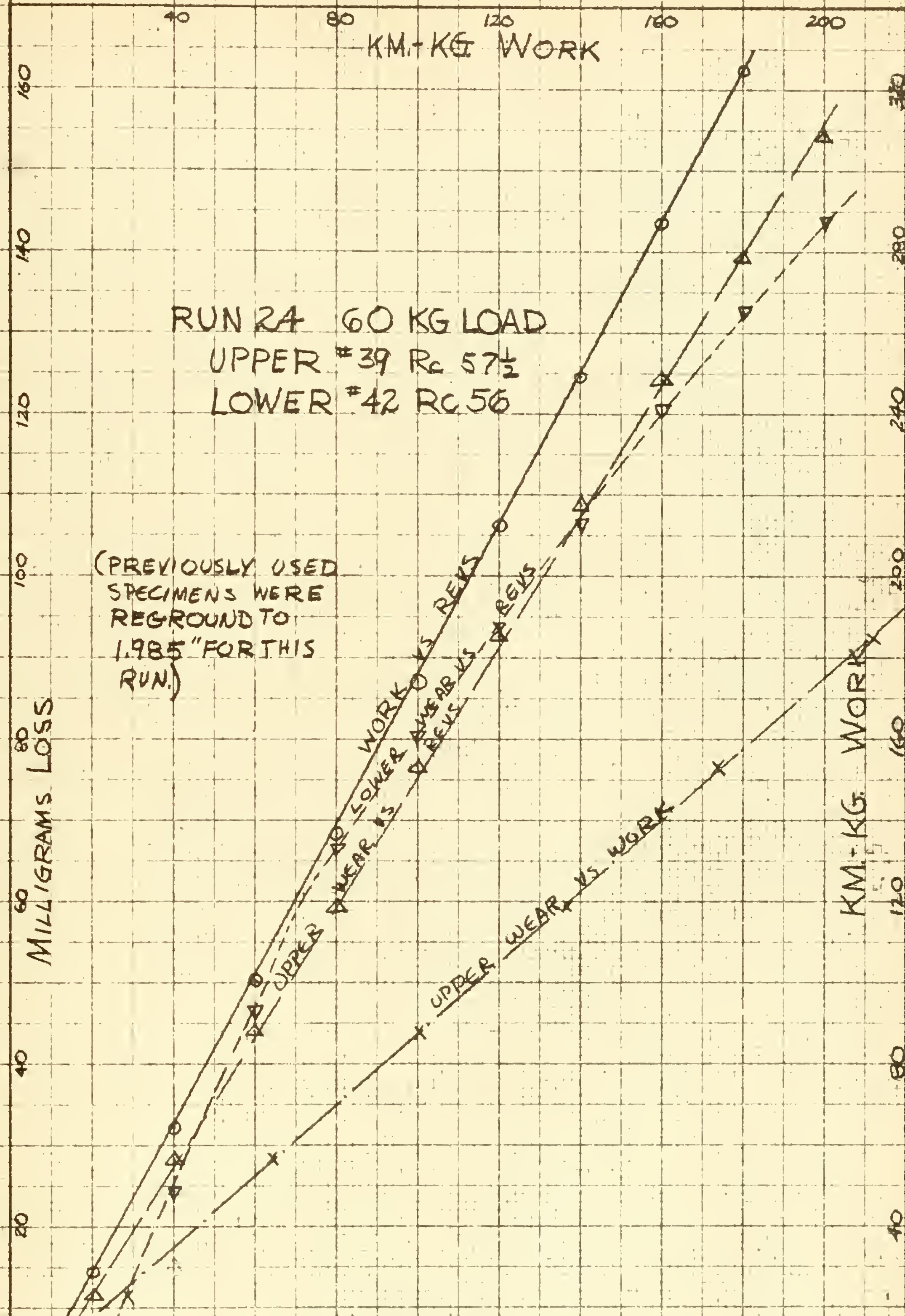


Fig. 31. Curves of results for Run 24 at 60 kg. load

APPENDIX IV

CALCULATION OF THE CONTACT STRESS

An equation for the maximum principle contact stress is given by Seely and Smith [17] for the case of two cylinders in line contact with a normal load only. This equation is

$$\sigma_z = -\frac{b}{\Delta}$$, where the z axis is normal to the surface of contact and where

$$b = \sqrt{\frac{2q\Delta}{\pi}}$$. Δ is a constant as indicated below.

Substituting for b in the above expression,

$$\sigma_z = -\frac{\sqrt{\frac{2q\Delta}{\pi}}}{\Delta}$$, and therefore $$\sigma_z = -\sqrt{\frac{2q}{\pi\Delta}}$$

q is defined as the normal load per unit length. For the case of identical cylinders

$$\Delta = 2r\frac{1-\nu^2}{E} = 2r\frac{(1-\nu)(1+\nu)}{E}$$ where E and ν are the elastic constants of the material and r is the cylinder radius.

The elastic constants of the material were unknown, and the below values of Young's modulus, E, and the modulus of rigidity, G, were chosen from a handbook [18] .

$$G = \frac{E}{2(1+\nu)}$$, and if E=29,500,000 psi. and G=11,500,000 psi.

$$\nu = 0.285.$$

If the radius of the cylinders is one inch,

$$\Delta = 2(1\,in.)\frac{(1-0.285)(1+0.285)}{29.5\times10^6\,psi.} = \frac{2(0.715)(1.285)}{29.5\times10^6}\ in.^3/lb.$$

$$\sigma_z = -\sqrt{\frac{2(29.5\times10^6)}{\pi(2)(0.715)(1.285}}\sqrt{q}$$

$$\sigma_z = -\sqrt{10.23\times10^6}\sqrt{q}$$

$$\sigma_z = -3.20\times10^3\sqrt{q}$$ psi. if q is in lbs./in.

Due to the axial reciprocation of the swing the load per unit length of cylinder, q, is not a constant for a given normal load. The length of the specimens used was 0.40 in., but at the points of greatest reciprocation of the swing the line of contact was only 0.242 in. long. Therefore, maximum and minimum values of the maximum principle stress must be found.

Where L is the normal load in pounds,

$$\sigma_{z\,max.} = -3.20\times10^3\sqrt{\frac{L}{0.242}} = \frac{-3.20\times10^3}{0.491}\sqrt{L} = -6.51\times10^3\sqrt{L}\ \text{psi.}$$

$$\sigma_{z\,min.} = -3.20\times10^3\sqrt{\frac{L}{0.40}} = -\frac{3.20\times10^3}{0.631}\sqrt{L} = -5.07\times10^3\sqrt{L}\ \text{psi.}$$

When the normal load is 60 kg.,

$$\sigma_{z\,max.} = -6.51\times10^3\sqrt{60\,\text{kg}\times2.205\ \text{lbs}/\text{kg.}} = -74{,}900\ \text{psi.}$$

$$\sigma_{z\,min} = -5.07\times10^3\sqrt{60\times2.205} = -58{,}200\ \text{psi.}$$

Applying this last calculation for other values of the normal load, one obtains:

L	$-\sigma_{z\,max}$	$-\sigma_{z\,min}$
50	68,300	53,200
40	61,100	47,600
30	52,900	41,200
20	43,200	33,700
14.5	36,800	28,700
4.5	20,500	16,000

The above values are in close agreement with those calculated by Rosenberg [11] . However, it must be stated again for emphasis that these stresses are for the case of normal load only. In the case of the test specimens, the axial reciprocation would introduce an axial frictional force in addition to the tangential frictional force. These

Frictional forces combined with the normal force present a three dimensional problem the solution of which was not attempted.

SE 23 57
FE 7 61
SE 4 62

4 5 4 0
1 0 4 3 6
1 1 4 1 0

Thesis
D718 Doxey 28802
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